S·A·E JOURNAL

PUBLISHED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.

D. G. Roos, President — David Beecroft, Treasurer

John A. C. Warner, Secretary and General Manager

Norman G. Shidle, Executive Editor

Vol. 35 — SEPTEMBER, 1934 — No. 3

CONTENTS

Publication Office, 56th and Chestnut Sts., Philadelphia, Pa.; Editorial and Advertising Departments at the headquarters of the Society, 29 West 39th St., New York, N. Y. Western Advertising Office, Room 2-136 General Motors Bldg., Detroit, Mich.

50,000 Car-Miles Run in 1934 Tests of Fuel Detonating Characteristics

By C. B. Veal
Research Manager, Society of Automotive Engineers

A TOTAL of 50,000 car-miles was involved in the test runnings at Uniontown, Pa., in the 1934 Cooperative Road Tests which came to a close early in August. This investigation was designed to check the applicability to present service conditions of the C.F.R. Motor Method (A.S.T.M. designation D 357-33 T) for determining the detonation characteristics of fuels, and to indicate a path for future effective antiknock research. It was carried out under the auspices of the Cooperative Fuel-Research Committee, with the participation of 25 organizations representing the oil and automotive industries in this and three other countries.

During these tests, 2516 gal. of test and reference fuels were rated in 19 cars including the 1934 models of 13 different makes. To achieve the desired results, 41 technical specialists toiled continuously through a month of perfect outdoor testing weather, working close to 10 hr. a day.

Drivers and observers, riding sometimes under conditions of extreme discomfort with all windows closed to exclude noises other than the fuel knock they were straining to detect accurately, covered an average of over 200 miles per day of strenuous running, up and down the steep incline of the Uniontown Hill.

A group of mechanics, aided by such equipment as a power analyzer and exhaust-gas analyzer, and, most important of all, by their own sure skill, maintained the cars in perfect test adjustment and running condition, in spite of the grueling service to which they were subjected.

Men assigned to the fuel shed carried on the painstaking job of blending and apportioning fuels to the test cars.

A fourth group, charged with the tabulation of the data and the planning of assignments in accordance with the conditions revealed by such data, often worked far into the night so that these assignments might be ready for an early morning start on the part of the test drivers and observers.

New features of this second series of Cooperative Road Tests, not included in the similar project two years ago, were: (1) level road runs; (2) so-called "blind tests"; (3) an investigation of the effect of the type of reference fuel on road knock ratings; (4) variations in spark settings; (5) blending of non-knocking test fuels with a low octane gasoline to obtain a rating by extrapolation; (6) observations with the "electric ear"; and (7) high speed cross country runs.

Personnel of 1934 Cooperative Road Tests

An Operations Committee, appointed by T. A. Boyd, chairman of the Subcommittee on Methods of Measuring Detonation of the Cooperative Fuel Research Steering Committee, exercised executive supervision over the carrying out of the test program. The membership of this committee follows:

Operations Committee

H. F. Huf, Chairman, The Atlantic Refining Co.
D. P. Barnard, Standard Oil Co. (Indiana.)
A. E. Becker, Standard Oil Development Co.
T. A. Boyd, General Motors Corp., Research Division.
*B. A. Kulason, The Texas Co.
L. C. Lichty, Yale University.
J. B. Macauley, Chrysler Corp.
T. B. Rendel, Shell Petroleum Corp.
*C. H. Sprake, Anglo-American Oil Co.
R. Stansfield, Anglo-Persian Oil Co., Ltd.
C. B. Veal, Secy., Society of Automotive Engineers, Inc.
C. R. Wagner, The Pure Oil Co.

* Drafted to membership by the Committee to assist in its work.

The upper stretch of Laurel Hill, near Uniontown, Pa., better known as Uniontown Hill.

Members of the "Brain Trust" better known as the Operations Committee which directed the Uniontown Test Project.

Beginning at the left are R. Stansfield, Anglo-Persian Oil Co., London; C. B. Kass, Standard Oil Development Co.; A. B. Wilder, Pure Oil Co.; Miss Geneva Prato, acting secretary of the Committee; C. B. Veal, research manager of the Society of Automotive Engineers; W. M. Holaday, Standard Oil Co. (Indiana); H. W. Best, Yale University; J. M. Campbell, General Motors Corp.; Gilbert Way, Chrysler Corp.; and (with his back to the camera) L. E. Hebl, Shell Petroleum Corp.

Following are listed those actively engaged in the test work at Uniontown for the entire duration of the series under the designation, Test Corps. The first man listed for each company was the official representative of his company. J. M. Campbell was chosen as vice-chairman of this Committee and acted in the chairman's absence.

Among these were: Dr. D. P. Barnard, Standard Oil Co. (Indiana); Dr. A. E. Becker, Standard Oil Development Co.; Major D. G. Brandt, Doherty Research Co.; Dr. C. Bonnier, Station Nationale de Recherches et d'Experiences Technique, Office National des Combustibles Liquides of France; Major N. Champsour, Air Attache, French Embassy; H. K. Cum-

Test Corps

Company	*Name*
Anglo-American Oil Co. (London)	C. H. Sprake
Anglo-Persian Oil Co. (London)	R. Stansfield
Atlantic Refining Co.	*J. R. Sabina
	N. Walton
Chrysler Corp.	*Gilbert Way
	Harold Evans
Doherty Research Corp.	George J. Liddell
	R. C. Porter
	H. G. Koch
Ethyl Gasoline Corp.	Cleveland Walcutt
	Robert Mescher
General Motors Corp.	*J. M. Campbell
	A. W. Macfarlane
Imperial Oil Co., Ltd.	A. L. Helliwell
National Bureau of Standards	Clarence S. Bruce
Phillips Petroleum Co.	Harold N. Trimble
	Howard N. Evans
Pure Oil Co.	*A. B. Wilder
Shell Petroleum Corp.	*L. E. Hebl
	R. J. Greenshields
Sinclair Refining Co.	W. G. Ainsley
	L. E. Baker
Society of Automotive Engineers, Inc.	C. B. Veal
Socony-Vacuum Corp.	G. A. Hope
	W. S. Mount
Standard Oil Co. of Calif.	J. R. MacGregor
Standard Oil Development Co.	*C. B. Kass
	J. Bouma
Standard Oil Co. (Indiana)	*W. M. Holaday
	Manning Mack
Sun Oil Co.	A. L. Clayden
	L. S. Hawthorne
	N. B. Sargent
	L. G. Strohm
The Texas Co.	B. A. Kulason
	J. G. Cook
	K. L. Hollister
Tide Water Oil Co.	L. Raymond
Universal Oil Products Co.	W. H. Hubner
Waukesha Motor Co.	A. W. Pope, Jr.
Yale University	*H. W. Best

* Acted on behalf of absent member of organization as member of Operations Committee.

Much credit for the success of the tests must also be attributed to members of the C.F.R. Committee and its subcommittees and interested engineers and technologists, who, unable to be present for the entire month, ably assisted the Test Corps by comments and advice given on occasional visits.

mings, Bureau of Standards; Dr. H. C. Dickinson, Bureau of Standards; Dr. R. Haskell, The Texas Co.; H. F. Huf, Atlantic Refining Co.; Prof. L. C. Lichty, Yale University; E. F. Lowe, Society of Automotive Engineers, Inc.; Neil MacCoull, The Texas Co.; T. B. Rendel, Shell Petroleum Co.; C. R.

Wagner, The Pure Oil Co., and John A. C. Warner, Society of Automotive Engineers, Inc.

A group of 53 technical men highly expert in the question of detonation measurement heard the final report to the C.F.R. Detonation Subcommittee on the results of the tests, given on Aug. 8. Among these, in addition to those already mentioned, were: Earl Bartholomew, Ethyl Gasoline Corp.; T. A. Boyd, General Motors Corp., Research Division; Dr. Graham Edgar, Ethyl Gasoline Corp.; J. B. MacCauley, Chrysler Corp.; Gordon McIntyre, Imperial Oil Co.; E. B. Phillips, Sinclair Refining Co., and Dr. C. H. Schlesman, Socony-Vacuum Co.

Origin of 1934 Cooperative Road Tests

The 1934 Cooperative Road Tests were a supplement to and logical outgrowth of the similar project involving the correlation of laboratory and road knock ratings carried out two years ago.

Both the oil and automotive industries generously cooperated in carrying out this first series of road ratings at the Uniontown Hill in August, 1932. A road test method was developed and the ratings obtained by it were compared with those obtained by the laboratory method then in use.

This comparison indicated a need for modifying the laboratory method so that results from its use would correspond more nearly with those characteristics of actual usage indicated by the road test. Such modifications were incorporated in the C.F.R. Motor Method. This modified method was subsequently adopted by the American Society for Testing Materials as Tentative Method of Test for Knock Characteristics of Motor Fuels, A.S.T.M. designation D357-33T. The original laboratory method was retained as the C.F.R. Research Method because of its value from an experimental standpoint.

That the C.F.R. Motor Method might require revision from time to time as changes in engine design and fuel characteristics outmoded its provisions was definitely recognized. At the meeting of the Cooperative Fuel Research Committee held following the first series of road tests, on Sept. 12, 1933, a recommendation, proposed by the Detonation Subcommittee, that such revision be considered annually, was unanimously approved.

The desirability of conducting further road tests with current car models during 1934 was discussed at the October, 1933, meeting of the Detonation Subcommittee and at the meeting of the C.F.R. Steering Committee a month later. The Detonation Subcommittee voted favorably on the desirability of such tests at its Jan. 26, 1934, meeting and at four subsequent meetings drew up, discussed and approved a program for the proposed new series of road ratings. At its May 1, 1934, meeting the C.F.R. Steering Committee authorized the carrying out of this program.

General Plan of Tests

As stated in the approved program of the 1934 Cooperative Road Tests, their object was:

1. To check the validity of correlation between road knock ratings and laboratory knock ratings;
2. To indicate the path of research aimed at adapting fuels and engines to better advantage.

Participants were to include any organization in the United States willing to share both the work and expense involved in providing cars, equipment and supplies, together with representatives of foreign countries. Accordingly, invitations were extended to companies in the oil, automotive, motor-truck engine and allied industries. Companies participating, shown in the Test Corps list, provided the fuel, cars, sup-

Members of the Detonation Subcommittee which met at Uniontown, Aug. 8 sat with the personnel of the Test Corps to get a nearly complete picture of those directly involved in the Uniontown project.

In the picture reading from upper left to right are: L. E. Baker, Sinclair Refining Co.; A. L. Foster, *National Petroleum News;* R. Mescher, Ethyl Gasoline Corp.; E. Bartholomew, Ethyl Gasoline Corp.; W. G. Ainsley, Sinclair Refining Co.; A. W. Macfarlane, General Motors Corp.; D. G. Brandt, Doherty Research Co.; Leo L. Heyn, Manager, Summit Hotel; H. F. Huf, Atlantic Refining Co.; D. P. Barnard, Standard Oil Co. (Ind.); C. R. Wagner, Pure Oil Co.; H. K. Cummings, National Bureau of Standards; H. W. Best, Yale University; C. H. Schlesman, Socony-Vacuum Corp.; W. S. Mount, Socony-Vacuum Corp.; H. L. Koch, Doherty Research Co.; E. B. Phillips, Sinclair Refining Co.; C. Walcutt, Ethyl Gasoline Corp.; Second Row: A. L. Clayden, Sun Oil Co.; T. A. Boyd, General Motors Corp.; A. E. Becker, Standard Oil Development Co.; H. M. Trimble, Phillips Petroleum Co.; L. Raymond; Tide Water Oil Co.; H. A. Evans, Phillips Petroleum Co.; H. B. Evans, Chrysler Corp.; G. McIntyre, Imperial Oil Refineries; A. B. Wilder, Pure Oil Co.; J. R MacGregor, Standard Oil Co. of Calif.; Mrs. MacGregor; Mrs. Ainsley; Miss Geneva Prato; Mrs. Best; Mrs. A. L. Beall; Mrs. Walcutt. Third Row: K. L. Hollister, The Texas Co.; H. C. Dickinson, National Bureau of Standards; R. Stansfield, Anglo-Persian Oil Co. (London); R. C. Porter, Doherty Research Co.; L. E. Hebl, Shell Petroleum Corp.; J. R. Sabina, Atlantic Refining Co.; N. MacCoull, The Texas Co.; G. J. Liddell, Doherty Research Co.; Mrs. Liddell; Mrs. Helliwell; Mrs. G. Way; Mrs. Huf; Gilbert Way, Chrysler Corp.; J. B. Macauley, Chrysler Corp.; C. B. Veal, Society of Automotive Engineers, Inc. Fourth Row: G. H. Hope, Socony-Vacuum Corp.; A. L. Helliwell, Imperial Oil Refineries; B. A. Kulason, The Texas Co.; W. H. Hubner, Universal Oil Products Co.; C. B. Kass, Standard Oil Development Co.; G. Edgar, Ethyl Gasoline Corp.; R. J. Greenshields, Shell Petroleum Corp.; J. G. Cook, The Texas Co.; C. H. Sprake, Anglo-American Oil Co. (London); L. C. Lichty, Yale University; W. M. Holaday, Standard Oil Co. (Ind.); J. M. Campbell, General Motors Corp.; J. Bouma, Standard Oil Development Co.; C. S. Bruce, National Bureau of Standards; A. L. Beall, Wright Aeronautical Corp.

plies, and, most important of all, the personnel for carrying out the test program. They also each contributed $100 to a fund from which were met the expenses of one representative each of Yale University, the Bureau of Standards and the Society of Automotive Engineers, together with certain miscellaneous costs.

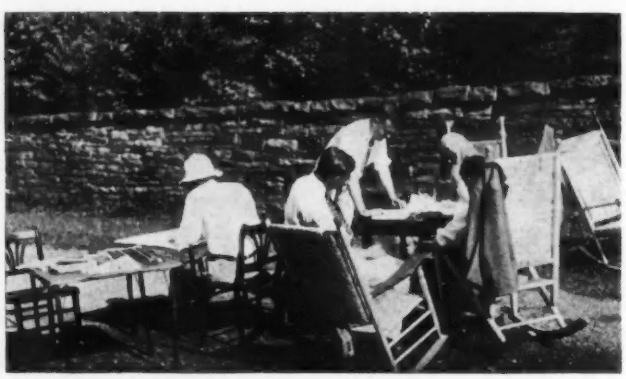
Invitations to participate were also extended to the Office National des Combustibles Liquides of France, to the Institution of Petroleum Technologists of Great Britain, and to the National Research Council of Canada. In response to these invitations, Monsieur C. Bonnier of the Station Nationale de Recherches et d'Experiences Techniques was appointed as the official French representative. Together with Major N. Champsour of the French Embassy, he devoted to observations of the tests a generous allotment of an exceedingly crowded stay in this country. C. H. Sprake of the Anglo-American Oil Co., Ltd., and R. S. Stansfield of the Anglo-Persian Oil Co., Ltd., were the emissaries of the British organization, while A. L. Helliwell of the Imperial Oil Co., Ltd., was present on behalf of the Canadian interests. The three last named actively cooperated in the carrying out of the entire test program.

The Operations Committee arranged to have the test fuels to be used in the road runs rated in the 19 C.F.R. cooperating laboratories according to the C.F.R. Motor Method (A.S.T.M. designation D357-33T) and the C.F.R. Research Method, so that laboratory determinations might be available with which to compare the road test results.

Procedure at Uniontown

The 41 men forming the permanent personnel of the Test Corps gathered at Uniontown on July 9. They were divided, according to specifications made by their companies and according to assignments of the Operations Committee, into 5 groups: (1) observers for making knock ratings; (2) drivers of the test cars; (3) fuel blenders; (4) maintenance mechanics and (5) the Operations Committee itself which made assignments and plotted the data as it became available. However, opportunity was given to engage in activity other than that designated in the original assignment, so that all who desired might have experience in the various types of job involved in an experimental investigation of this nature.

All cars used were 1934 models. They included 3 Chevrolets, 3 Fords and 3 Plymouths, and one each of the following makes: Buick, Cadillac, Chrysler, Dodge, Graham, Oldsmobile, Packard, Pontiac, Studebaker and Terraplane. This selection of 3 each of 3 makes in the popular-priced field and



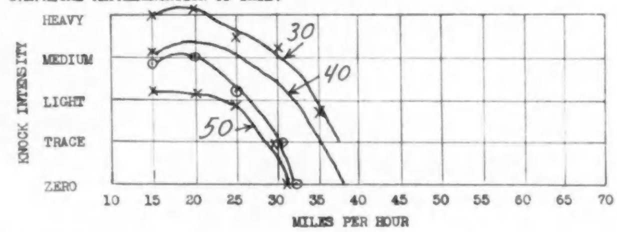
A segment of the Operations Committee at work showing various ideas about how to avoid the effects of the intense heat which made many of the tests a real hardship to the operators.

C.F.R. ROAD TESTS

CAR ______ NO. 41 ODOMETER ______ SHEET NO. 1
DATE 7-16 AIR ______ BAROMETER ______ HUMIDITY ______
TEST CONDITIONS HILL COMPRESSION RATIO ______ SPARK STANDARD
OBSERVER JOHN DOE GROUP 125

FUEL	MILES PER HOUR 10	15	20	25	30	35	40	45	50	55	60	65	70	WATER TEMP.
50-C		L+	L+	L	T	O		33						160
50-C		L+	L+	L+	T	O		35						165
40-C		M	M+	M	L	T	O	40						160
40-C		M	M	M-	L-	T	O	36						165
30-C		H	H	H-	H-	L+	T							160
40-C		M+	H	M	M	T	O	38						160
40-C		M	M	M	L	T	O	37						165
#54		M	M+	L+	T	O		34						160
#54		M-	M-	L	T	O		31						165

GRAPHICAL REPRESENTATION OF DATA:



SUMMARY:

FUEL	RATING	REMARKS
#54	43-C	

Fig. 1—Log Sheet on Which Each Observer Recorded His Ratings

one each of 10 more expensive makes was thought to approximate the distribution of cars in use, as well as could be done with the limited number of cars feasible for the test program. Of these the following had optional high compression heads: Chrysler, Dodge, Plymouth and Terraplane.

The supply of test fuels consisted of 300 gal. each of 7 commercially-marketed, branded fuels and 250 gal. each of 9 unbranded fuels chosen to represent types as follows:

b-1	Vapor phase—600 gal.—70 octane number	Pure Oil Co.
b-2	Liquid phase—800 gal.—68 octane number	Standard Oil Co. (Ind.)
b-3	A-7 plus b-1 for 65 octane number	To be blended at Chicago by Standard Oil Co. (Ind.), Sinclair and Pure Oil and shipped to Uniontown.
b-4	A-7 plus b-2 for 65 octane number	
b-5	b-3 plus T.E.L. for 70 octane number	
b-6	b-4 plus T.E.L. for 70 octane number	
b-7	b-2 plus T.E.L. for 75 octane number	
b-8	A-7 plus benzol for 70 octane number	
b-9	Straight run California (1600 gal.) 71 octane number	Standard Oil Co. (Calif.)

The identity of the branded fuels was concealed throughout the tests. The identity of the non-branded fuels was revealed at the completion of specific portions of the test, but these fuels were recoded for the remainder of the tests. For reference fuels, Standard Oil of New Jersey Reference Fuels C-8 and A-3 were blended in varying percentages. Where

reference fuels of higher octane number than the high reference fuel were required, the high reference fuel plus tetraethyl lead was used.

Discretion was left to the Test Group as a whole, under the direction of the Operations Committee, as to certain details of methods to be followed. The approved program provided that the C.F.R. Road Test Method for conducting antiknock tests on motor gasolines as described in a previous issue of the S.A.E. Journal should be used.[1] However, it also gave general directions that additional acceleration tests on level road and at high speed should be run, and also that other procedures should be investigated.

In general, the basic routine testing, serving as a foundation for the series of tests and as a starting point for variations was done in the method already worked out at the 1932 series and then found to be effective.

Record and Analysis Forms

In practice, each observer recorded his ratings on the Log Sheet shown in Fig. 1. He determined for the test fuel the

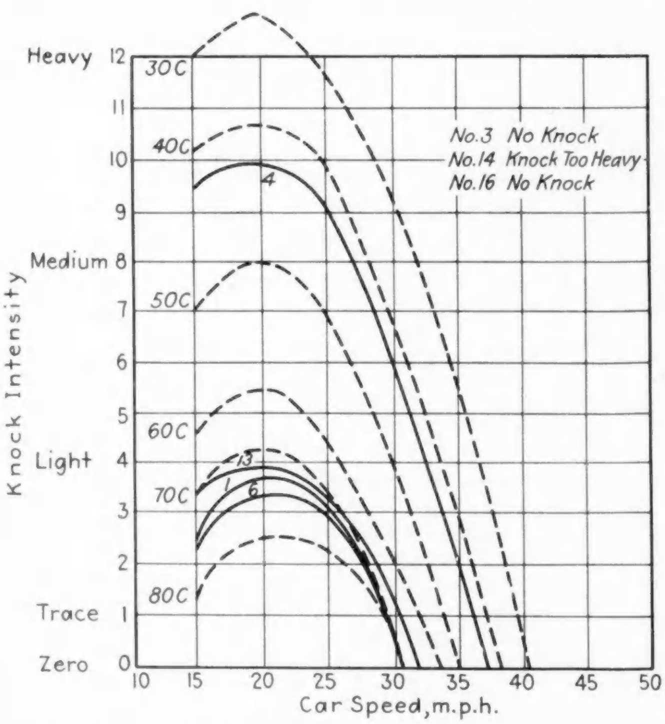
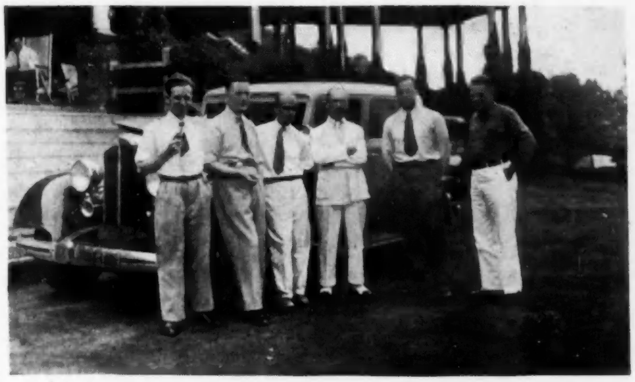


Fig. 2—Sample of Cumulative Chart Incorporating All Findings for All Branded Fuels in One Particular Car

degree of knock—heavy, medium, light, trace or zero—at speed increments of 5 m.p.h., and then ascertained what blends of the reference fuels C-8 and A-3 would be required in the car being used, to give, first, a slightly heavier knock than that produced by the test fuel, and second, a slightly lighter knock. The rating of the test fuel expressed as the per cent of C-8 in A-3 would lie between these two. As is shown on the Log Sheet, not only the test conditions and engine adjustments were noted for each run, but also the atmospheric conditions, full weather reports being kept.

These data were turned into the group in charge of charting and analysis, which functioned continuously and kept pace with the test determinations in its work. Two types of cumulative chart were prepared, as shown in Figs. 2 and 3.

[1]See S.A.E. Journal, March, 1933, p. 105.



Several representatives for foreign automotive and petroleum interests participated in the tests.

Left to right: R. Stansfield, Anglo-Persian Oil Co., London; C. H. Sprake, Anglo-American Oil Co., London; A. L. Helliwell, Canada; Major N. Champsour, Air Attache, French Embassy; Dr. C. Bonnier, Station Nationale de Recherches et d'Experiences Technique, Office National des Combustibles Liquides of France; and the author, C. B. Veal.

One of these, shown in Fig. 2, incorporated all the findings for all the branded fuels in one particular car. These were known as "Framework Charts" because the knock ratings at various speeds for the *standard* blends of reference fuel were first charted on them, as shown by the dotted lines in the fictitious examples given. The framework was then filled in with ratings for the various test fuels which could then be compared not only with the reference fuels but with each other. In the hypothetical car represented in this chart, four fuels were found to knock with intensities lying between those of reference fuels 40 per cent of C-8 and 80 per cent of C-8; two fuels did not knock at all and one knocked so heavily that it could not be rated within the range of reference fuels used.

In the second type of cumulative chart, shown in Fig. 3, the ratings for an individual fuel in all cars were shown. As is illustrated in this fictitious case, at least two determinations and sometimes three were made for each fuel in each car.

Variations from Standard Procedure

One variation from standard procedure is also indicated in Fig. 3, that is, the change in the degree of spark advance from the basic factory spark setting. Additional rules or changes in standard procedure were adopted for three reasons: first, to increase the accuracy of results; second, to secure knock where no knock was obtainable with the use of the

(Concluded on page 28)

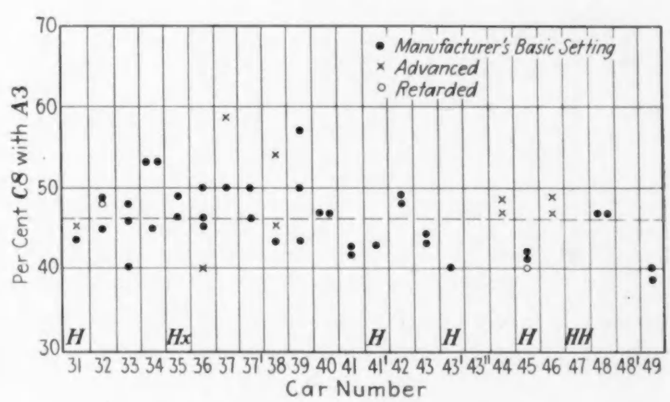
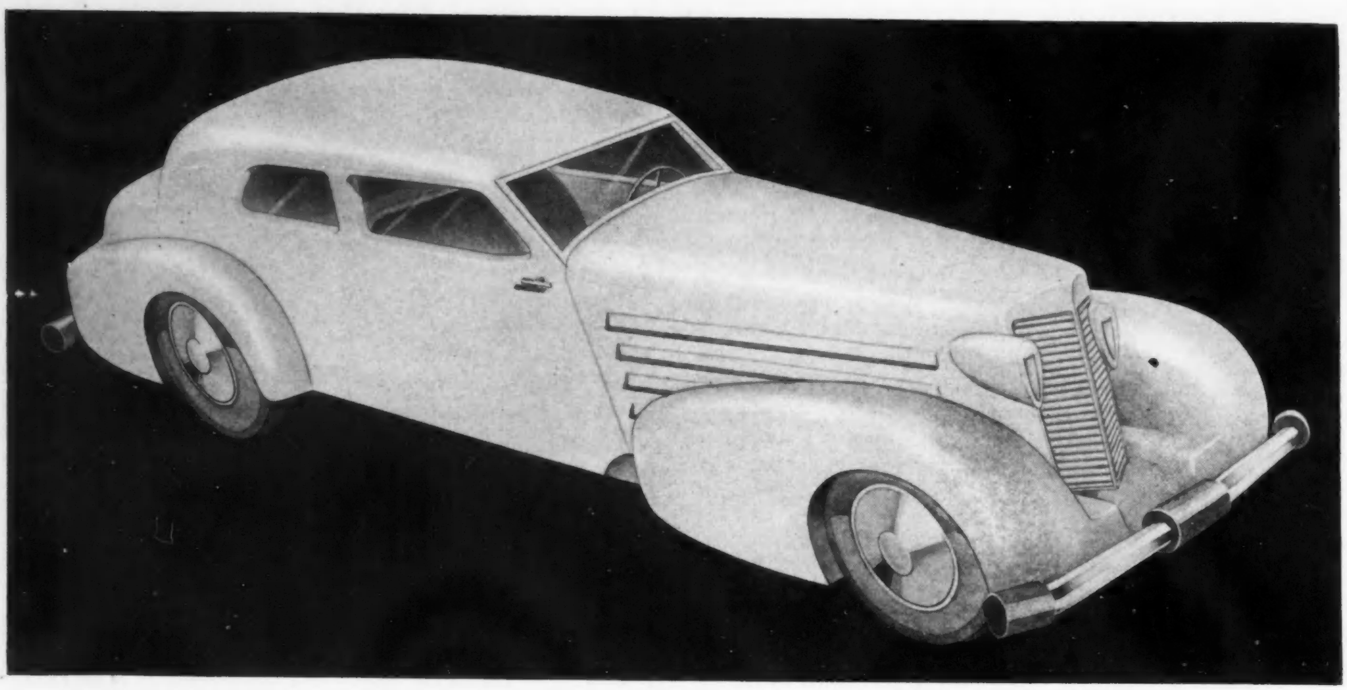


Fig. 3—Another Type of Cumulative Chart Showing Rating for an Individual Fuel in All Cars



Basic Principles of Body Design Arise from Universal Rules

By Walter Dorwin Teague

THE basic laws of design are unchangeable, invariable. They derive their validity from the structure of this universe and the structure of our perceiving minds.

The student of design needs to be firmly grounded in these abstract basic laws if he is to be able to analyze his problem, whatever it may be, and work out a sound solution. The soundness of his taste and the certainty of his insight will be enormously strengthened if he has studied design not in one special field alone, but in many diverse and apparently unrelated phases. If he is able actually to see the same laws governing the design of a Greek temple, a Gothic cathedral, and a late Victorian carriage, and at the same time understand why these laws produce such widely different results, he can approach the design of such a highly specialized product as an automobile with far more competence and confidence.

The inquiring student soon sees that the application of the laws of design varies inevitably, incessantly. Their application *must* vary, because the first law of all designs is *fitness*.

This principle of fitness may be otherwise stated as perfect adaptation of means to end. The design of any object, natural or man made, is inherent in the object itself, and must be evolved inevitably out of the function which the object is adapted to perform, the materials out of which it is made, and the methods by which it is made. The designer is not an inventor trying to create new and unprecedented forms; he is an explorer seeking the one perfect form concealed within the object beneath his hand. He does not *apply* design—he *evokes it out of* the subject of his labors.

William Morris used to say that the useful is always beautiful. This is not strictly true, because we make good use sometimes of very poor tools. What I think Morris meant is that when a thing is perfectly adapted to the function it is intended to perform, when its ultimate form has been achieved, that thing is beautiful.

When the engineer designs purely for practical engineering ends, he produces beautiful results just in so far as he is successful in solving his problem; it is only when he strays from pure engineering and begins to design in a conscious effort to improve on the bare solution of his problem that he goes wrong. That is why airplanes and speed boats are so much more beautiful than automobiles: in the first two instances the designer stuck to his job and admitted nothing into his design which did not contribute to its performance; in the third he has been deflected from his simple aim by tradition, fashion and a dozen other irrelevant influences, and has wandered off after strange gods. Curiously enough, the artists who are invading industry today are doing so for the purpose of putting engineering back on its own track, inducing engineers to build for the public in the same way that they build for purely engineering ends.

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

The picture at the top of this page is a special sports body designed for Col. Howard Marmon by Mr. Teague. Body completed in April, 1933, anticipating certain 1934 styles.

Thus beauty, of which we hear so much, is not the aim of design: beauty is simply the visible evidence that design has been successful. The aim of design is a perfectly functioning organism, and objects are beautiful just in so far as they function perfectly: this is true of race horses, panthers, oak trees, and it is true of swords and ox carts, airplanes and motor cars.

Fitness is a threefold problem, involving not only function but materials and construction as well. In two different design problems, two of these determining elements may be the same, but a difference in the third will produce a wholly different result.

The first step in design, then, must be a complete and adequate definition of function, a study of the nature and potentialities of materials, a familiarity with the technologies of construction. These are not always easy to master: the function of a newly invented product such as an automobile may not reveal itself at once in its entirety, materials can only be understood through experiment, technologies develop and change as they are practised. Hence the ultimate form of an object may forever elude us, and the finest results may be the last to be achieved before the object itself is obsoleted and superseded by some wholly new form; as the last carriages to be built were the finest.

When internal combustion engines superseded horses as a means of propelling vehicles, very few of the implications of this change were realized. Speed requirements have steadily risen, along with standards of passenger comfort; improved roads, improved motors, new materials and new methods of fabrication have steadily advanced our objective. Therefore, the designers of ten years ago are not to be ridiculed because they did not produce the automobile of today: the automobile of today was then still in the egg. The designers of any time are to be criticized only in so far as they fail to use with relentless logic all the knowledge and all the means they possess: by this standard I'm afraid we are as much at fault today as ever we were.

As we study function, materials, construction, and seek to evolve out of them a perfect form, we must be guided always by certain universal principles which we find are exemplified in all good design. First of these is *Unity*.

Any organism must be conceived as a unity, one theme, one purpose, must dominate it; all its elements must be integrated as closely as possible, so that it looks as if it had been poured in a single mold.

Automobile design has struggled always to free itself from irrelevant detail and the battle is far from won: it has struggled to achieve an integration of its forms, and in this it has made more progress in the past four years than in the preceding ten. But its advance will be hastened when designers no longer grope toward unity, but seek it consciously and intelligently.

Related to unity as means to end is the ideal of *Simplicity*. In design, no arrangement, no form, can be right if a simpler and more direct arrangement is equally efficient. The designer will keep himself out of no end of traps if he will make the word "Simplify" his motto; if he will criticize his own work constantly in a search for unnecessary curves and complexities, non-essential gadgets and adornments.

One's development of a simple, unified form must always be guided by a sense of *Proportion*.

Proportion is the means whereby one establishes harmonious relationships between the elements of a form, and corresponds to harmony in music. As in music, harmonious relationships in design are governed by mathematical laws, but no amount of mathematics will take the place of a good ear or a good eye.

Closely allied with a sense of proportion is a sense of *Line*.

All material objects may be seen in silhouette, and so complicated a form as a motor car is traversed by many lines within its silhouette. The quality of all these lines may be pleasant or the reverse, but unless the line is right the eye again will not be satisfied. It is almost impossible to define in words the quality of a line which makes it right, since that quality varies with position and relationships. But it may be cited as an example that a few years ago almost all fender lines were bad; today they are mostly fairly good, because they have been studied intelligently; but the rear lines of many cars today, as seen in silhouette, are definitely bad.

Over and beyond these factors there is, of course, the innate ability of the designer, which enables him to conceive fresh and unexpected forms and arrangements, imparts charm and grace to his work, gives it that plus quality which can never be obtained by rule of thumb. But nothing the ablest designer does—except when he errs—will be done in violation of these basic principles; and the designer who lets himself be deflected from them by any consideration whatosever, is simply falling into error.

The question will be raised, quite rightly: "We build motor cars to sell; suppose these principles aren't enough to satisfy the public?" The answer is this: it can be shown conclusively that every success in the automobile industry has resulted from an approach toward a realization of some one or other of these principles. No one has yet approached them all, since the ultimate motor car has not yet been built; but the industry has steadily progressed toward their realization. And as they are embodied in cars, the public response is as inevitable as it is to Gallant Sir, Johnny Weismuller, or Dolores del Rio; all of whom are built on these principles.

Walter Dorwin Teague

News of the Sections

S. A. E. Sections Elect Officers For 1934-5 Administrative Year

● Baltimore

Chairman: Charles Froesch, sales and service engineer, General Aviation Corp.; vice-chairman: Major James R. Hill, assistant commandant, Q.M.C. Motor Transport School, U. S. Army, Holabird Q.M. Depot; treasurer: Robert C. Hall, engineering aide, United Railways and Electric Co.; secretary: Espy W. H. Williams, statistician, Automotive Trades Alliance.

Charles Froesch

L. P. Saunders

● Buffalo

Chairman: Laurence P. Saunders, director of engineering, Harrison Radiator Co.; vice-chairman: Karl M. Wise, director of engineering, Pierce-Arrow Motor Car Co.; secretary-treasurer: Walter R. Ramsaur, research engineer, Harrison Division of General Motors.

J. C. Armer

● Canadian

Chairman: J. C. Armer, vice-president, Dominion Forge and Stamping Co., Ltd.; vice-chairman: N. P. Petersen, manager, Canadian Acme Screw & Gear, Ltd.; treasurer: Marcus L. Brown, factory manager, Seiberling Rubber Co. of Canada, Ltd.; secretary: Warren B. Hastings, editor, manager, *Canadian Motorist.*

● Chicago

Chairman: R. E. Wilkin, Standard Oil Co. of Indiana; vice-chairman: Robert T. Hendrickson, treasurer, sales manager, Hendrickson Motor Truck Co.; treasurer: H. F. Bryan, carbureter engineer, International Harvester Co.;

R. E. Wilkin

secretary: R. B. May, curator, automotive engineer, Museum of Science and Industry.

● Cleveland

Chairman: A. K. Brumbaugh, Cleveland Club; vice-chairman: A. T. Colwell, chief engineer, Thompson Products, Inc.; vice-chairman, representing Akron and Canton members: K. D. Smith, technical superintendent, Tire Division, B. F. Goodrich Rubber Co.; treasurer: Ralph C. Chesnutt, experimental engineer, Cleveland Tractor Co.; secretary: Dr. Paul E. Hemke, associate professor, Department of Mechanics, Case School of Applied Science.

● Dayton

Chairman: Frank G. Born, engineer, Delco Products Corp.; vice-chairman: Charles S. McCann, vice-president, Hydraulics Products Co.; treasurer: Charles L. Lee, chief engineer, Lee Engineering Products Corp.; secretary: Karl H. Glanton, vice-president in charge Mechanical Sales Division, Dayton Rubber Mfg. Co.

● Detroit

Chairman: C. R. Paton, chief engineer, Packard Motor Car Co.; vice-chairman for passenger-cars: E. H. Smith, development engineer, Packard Motor Car Co.; vice-chairman for bodies: E. E. Lundberg, chief engineer, Briggs Mfg. Co.; vice-chairman for aeronautics: Prof. Peter Altman, director, Aeronautic Department, professor aeronautical engineering, University of Detroit; vice-chairman for production: W. B. Hurley, sales engineer, Detroit Edison Co.; vice-chairman for students: O. E. Kurt, tire development engineer, U. S.

From Montreal on the East to Windsor on the West, members of the Canadian Section journeyed to the guests. The celebrated hospitality of the Canadian Section brought out some more people to dinner and

C. R. Paton

Tire Co., Inc.; treasurer: F. W. Marschner, western sales manager, New Departure Mfg. Co.; secretary: C. O. Richards, body engineer, Cadillac Motor Car Co.

● Indiana

Chairman: A. W. S. Herrington, president, Marmon-Herrington Co., Inc.; first vice-chairman: Herman E. Winkler, assistant chief engineer, Schwitzer-Cummins Co.; second vice-chairman: W. K. Creson, assistant chief engineer, Ross Gear & Tool Co.; third vice-chairman: Daniel C. Teetor, in charge of manufacturing, Perfect Circle Co.; treasurer: P. A. Watson, vice-president, Duesenberg, Inc.; secretary: Harlow Hyde, Indianapolis.

A. W. Herrington

● Kansas City

Chairman: S. M. McKee, superintendent, lubricants engineer, Nourse Oil Co.; vice-chairman: Morris Cohen, industrial engineer, Schulze Baking Co.; treasurer: F. C. Buchanan, director of sales, Columbia Steel Tank Co.; secretary: C. A. Shepard, vice-president, Inter-State Oil Co.

● Metropolitan

Chairman: Sid. G. Harris, Eastern manager, Burton Auto Spring Corp.; vice-chairman for aeronautics: Hector Alexander, pilot, chief tester, Cadillac Motor Car Co.; vice-chairman for marine: W. E. John, sales engineer, Hall-Scott Motor Car Co.; vice-chairman: A. E. Becker, Standard Oil Development Co.; treasurer: E. C. Blackman, sales engineer, Thermoid Rubber Co.; secretary: O. P. Liebreich, power brake engineer, Smith & Gregory of New York, Inc.

C. E. Frudden

Sid. G. Harris

● Milwaukee

Chairman: C. E. Frudden, chief engineer, West Allis Tractor Div., Allis-Chalmers Mfg. Co.; vice-chairman: C. M. Eason, secretary-treasurer, Fawick Mfg. Co.; treasurer: A. W. Pope, Jr., research engineer, Waukesha Motor Co.; secretary: R. W. Wilson, secretary-general manager, Perfex Corp.

Scarboro Golf Club in Toronto. The day was July 6. and the (golfing) attendance 148 members and all-in-all the 1934 outing of the Section was a distinct success. It's an annual affair, y'know.

C. E. Batstone

• New England

Chairman: Charles E. Batstone, manager of sales, International Harvester Co.; vice-chairman: Cyril C. Lawton, works manager, White & Bagley Co.; treasurer: Albert Lodge, owner, general manager, Albert Lodge, Inc.; secretary: Richard R. Whittingham, lubricants engineer, Standard Oil Co. of New York, Inc.

• Northern California

Chairman: George H. Mosel, Pacific Coast manager, Raybestos Division of Raybestos-Manhattan, Inc.; vice-chairman: Joseph F. Long, owner, J. F. Long Co.; treasurer: Ulysses A. Patchett, instructor, mechanical engineering, Stanford University; secretary: William S. Crowell, claims adjuster, Independent Indemnity Co.

• Northwest

Chairman: John G. Holmstrom, chief engineer, Kenworth Motor Truck Corp.; vice-chairman: James H. Frink, assistant to manager, Washington Iron Works; treasurer: Maurice J. Kane, Seattle; secretary: George E. Bock, Seattle.

J. G. Holmstrom

• Oregon

Chairman: Harley W. Drake, superintendent of garage, Portland Gas & Coke Co.; vice-chairman: E. H. Swayze, treasurer, Lineham Motor Corp.; treasurer: J. Verne Savage, shop superintendent, City of Portland Municipal Shop; secretary: Joseph P. Seghers, president, Seghers Motor Co.

H. W. Drake

• Philadelphia

Chairman: James B. Franks, Jr., sales manager, Philadelphia Branch, White Motor Co.; vice-chairman: J. C. Geniesse, research engineer, Atlantic Refining Co.; treasurer: L. M. De Turk, experimental and development engineer, J. G. Brill Co.; secretary: Joseph Geschelin, engineering editor, Chilton Co.

• Pittsburgh

Chairman: A. R. Platt, Pittsburgh; vice-chairman: F. W. Heisley, manager, Joseph Woodwell Co.; treasurer: J. J. McNally, salesman, White Co.; secretary: R. N. Austen, sales manager, Iron City Spring Co.

A. R. Platt

W. R. Brashear

• St. Louis

Chairman: W. R. Brashear, assistant manager, Aviation Dept., Shell Petroleum Corp.; vice-chairman: W. E. Ziegenbein, vice-president, Borbein-Young & Co.; treasurer: R. M. Pease, manager, St. Louis Factory, Axelson Aircraft & Service Co.; secretary: George R. Gwynne, sales representative, White Co.

• Southern California

Chairman: E. E. Tattersfield, president, Electric & Carbureter Engineering Co.; vice-chairman: R. N. Reinhard, automotive engineer, Western Dairy Products, Inc.; vice-chairman for aeronautics: C. F. Lienesch, technical representative, Union Oil Co. of California; treasurer: J. Jerome Canavan, partner, Canavan & Kunkel; secretary: W. E. Powelson, master mechanic, Fire Dept., Los Angeles County.

• Syracuse

Chairman: H. Follett Hodgkins, president, treasurer and general manager, W. C. Lipe, Inc.; vice-chairman: L. W. Moulton, partner, Manufacturers' Consulting Engineers; treasurer: R. F. Russell, tool supervisor, equipment engineer, American La France & Foamite Corp.; secretary: Rodman S. Reed, chief engineer, Brockway Motor Co., Inc.

Metropolitan Plans Annual "Get-Together"

• Metropolitan

The Metropolitan Section will hold a Summer Meeting at the Monmouth Hotel, Spring Lake, N. J., on Sept. 7, 8 and 9. To this annual "get-together" members of the Philadelphia and Baltimore Sections have also been invited. All other members of the Society, and their guests, will receive a welcome too.

There will be a broad program of sports. Golfers will compete for the Winchester cup, and a second cup has been offered for sports by the Standard Oil Co. of N. J.

Reservations for the meeting may be sent to the Metropolitan Section in care of S.A.E. headquarters, 29 West 39th Street, New York City.

Matthews Sees Denver Group

• Denver

Enroute to Kansas City from a vacation in Alaska, Ralph R. Matthews, sales manager, Battenfeld Grease & Oil Corp., stopped in Denver to give a talk on lubrication to the S.A.E. Club of Denver. The meeting was attended by 75 members and guests, and was held in the "run-down" room of the Motoroyal Oil Co. Dean M. Gillespie of the latter company heads the Denver group.

James F. Fox

• Washington

Chairman: James Fulton Fox, mechanical engineer, U. S. Navy Dept., Bureau of Ordnance; vice-chairman: Philip R. Wheeler, mechanical engineer, Navy Department; treasurer: Major Stephen G. Henry, Infantry (Tanks), U. S. Army, War Dept.; secretary: Clarence S. Bruce, assistant mechanical engineer, U. S. Bureau of Standards.

Armstrong Speaks On Rail Car Topic

• So. California

W. R. Armstrong, superintendent, Union Pacific Railroad, was the speaker at a meeting of the Southern California Section on June 8. Mr. Armstrong's subject was "Automotive-Powered Streamlined Railcars."

C. F. Lienesch, F. C. Patton, Ethelbert Favary, W. I. Hodge, C. V. Elliot and C. T. Austin participated in the discussion at the meeting, which was attended by 60 members and guests.

Treasurer is Host To Oregon Section

• Oregon

A Summer picnic and party, attended by 75, was arranged by the Oregon Section on July 14. The scene was the Oswego Lake home of J. Verne Savage, treasurer of the Section and superintendent of the Municipal Shops in Portland. The party began at four in the afternoon and lasted until 11. Fifty-six persons sat down to dinner, and more arrived for the boating, dancing and games which followed. The picture shows the lake-front yard of Mr. Savage's home.

Three-Session Program Completed For Production Meeting

WHEN the 1934 Production Activity Committee was doing its planning for the year, beginning last January, it based its plans on the possibility of a production session at the 1934 Summer meeting. Such a session did not prove practicable or desirable and therefore plans were laid for a general Production Meeting in Detroit on Oct. 10 and 11. The program for this meeting is virtually complete. The two-day meeting will be held at the Book-Cadillac Hotel with W. H. McCoy, vice-president representing the Production Activity, in charge.

The meeting will be climaxed by a production dinner at which the speaker of national prominence in automotive production circles will be featured. Clyde R. Paton, chief engineer of Packard Motor Car Co., and chairman of the Detroit Section, is cooperating in arranging the dinner, which will be sponsored jointly by the Detroit Section and the Production Activity.

At the Production Meeting a dynamic balancing session has been scheduled in order to make available the latest information on recent important developments in the field. Thomas C. VanDegrift and John M. Tyler of the General Motors Research Laboratories are to present at this session a paper titled "Balancing Problems in Automotive Engineering". Mr. McCoy, who is manager of the experimental production machine shop, General Motors Research Laboratories, will be in the chair.

At the session on broaching practice one paper will be devoted to a thorough review of up-to-date production experience with surface broaching. The intent of the paper will be to continue further discussion of earlier phases of the subject described by Joseph Geschelin at the Annual Meeting of the Society last January. E. S. Chapman of the Amplex Division of Chrysler will present the paper on "Production Experience with Surface Broaching". A study of costs with respect to purchasing production equipment will be presented at the same session by Joseph E. Padgett, factory manager of the Spicer Mfg. Corp. Joseph Geschelin, engineering editor of *Automotive Industries* and chairman of the S.A.E. Production Meeting Committee, will be in the chair at the broaching practice session.

On Wednesday afternoon, Oct. 10, there will be a cooperative cutting oil session under the direction of Chairman A. Ludlow Clayden, vice-president representing the Fuels and Lubricants Activity. A study of cutting lubricants and the fundamental characteristics in their selection for various types of metal-cutting jobs will be presented by W. D. Huffman, chief chemist of the Chevrolet Gear and Axle Plant in Detroit, in collaboration with C. B. Harding, Sun Oil Co., and W. H. Oldacre of D. A. Stuart & Co.

This paper is expected to throw a good deal of light on a subject which has been under discussion for several years particularly in the Standards Committee, to which has been suggested the possibility of establishing a standard or classification for cutting lubricants.

Program of S.A.E. Production Meeting Oct. 10-11

Following is the detail program for the Production Meeting, Oct. 10 and 11, Book-Cadillac Hotel, Detroit.

WEDNESDAY, OCT. 10

Afternoon — Cutting Oil Session — Chairman, A. L. Clayden

Paper—Cutting Lubricants and the Fundamental Characteristics in Their Selection

Author—W. B. Huffman, Chief Chemist, Chevrolet Gear and Axle Plant, Detroit

Collaborators—C. B. Harding, Sun Oil Co., Philadelphia

W. H. Oldacre, D. A. Stuart & Co., Ltd., Chicago

Evening—Dynamic Balancing Session—Chairman, W. H. McCoy.

Paper—Balancing Problems in Automotive Engineering

Authors—T. C. VanDegrift and J. M. Tyler, General Motors Research Laboratories, Detroit

THURSDAY, OCT. 11

Afternoon — Broaching Practice Session — Chairman, Joseph Geschelin

Papers—1. Production Experience with Surface Broaching

E. S. Chapman, Chrysler Motors, Amplex Division, Detroit

2. Study of Costs with Respect to Purchasing Production Equipment

J. E. Padgett, Spicer Mfg. Corp., Toledo, Ohio

Evening—Production Dinner—Sponsored jointly by Detroit Section

Toastmaster to be selected

Principal Speaker—To be announced later

M. V. Cost-Finding Report Available

THE complete report on the Uniform Motor-Vehicle Operating Cost Classification as finally approved by the Society during its Summer Meeting at Saranac Inn, New York, in June, is now available.

The general classification of breakdown of cost items, adopted by the Society some time ago, has been supplemented by a complete set of sample form headings and a grouping of these forms into three plans, one for the operators of a very small fleet, one for a moderate-sized fleet operator, and the third or complete plan for the large-sized operator.

Work on this project was first started by the Society at a meeting of the old Operation and Maintenance Activity Committee in New York in October, 1925. The cost record systems of a large number of motor-vehicle fleet operating companies were secured as the basis for the study and a Subcommittee appointed consisting of R. E. Plimpton, Chairman, who was then on the editorial staff of Bus Transportation; J. F. Winchester, Standard Oil Co. of New Jersey; B. V. Evans, Detroit Motorbus Co.; F. K. Glynn, American Telephone and Telegraph Co.; W. P. Kennedy, Kennedy Engineering Corp.; E. E. LaSchum, American Railway Express Co.; J. H. Lyman, City Transportation Co. (Tacoma); M. W. Glover, West Pennsylvania Railways, and F. C. Horner, General Motors Corp.

In June, 1927, a preliminary classification and definitions had been drafted which were the foundation of reports on this project that were published in the S.A.E. JOURNAL, issues of August and December, 1927, and February, 1928. Further studies of classifications were continued when in February, 1930, a new Subcommittee was appointed under the Transportation and Maintenance Activity which was the successor to the original Operation and Maintenance Activity of the Society. This Subcommittee consisted of F. C. Horner, Chairman, General Motors Corp.; J. F. Winchester, Vice-Chairman, Standard Oil Co. of New Jersey; W. M. Clark, S. S. Pierce Co.; F. K. Glynn, American Telephone and Telegraph Co.; Adrian Hughes, United Railways and Electric Co. of Baltimore, and E. S. Pardoe, Capital Traction Co.

By December, 1930, the Subcommittee submitted two classifications, one already referred to and one submitted by L. V. Newton, for a final selection between them, the fundamentals of which were published in the May, 1929, issue of the S.A.E. JOURNAL. The final report of the Transportation and Maintenance Activity Committee that was selected was referred to the Motorcoach and Motor-Truck Division of the S.A.E. Standards Committee in May, 1931, and formally adopted by the Society in June of that year.

This report was used by the Motor Transportation Committee and the General Records Committee of the National Electric Light Association as a basis for extending the classification to include a set of cost record forms that it was felt were necessary to make the classification of greatest practical use to the operators. The N.E.L.A. Committees of which J. M. Orr and M. C. Dowling were chairmen respectively, completed their report that was published as N.E.L.A. publication No. 221 in June, 1932. This work was again referred to the S.A.E. Transportation and Maintenance Activity Committee at the suggestion of L. V. Newton and J. M. Orr with the intention of making the S.A.E. Recommended Practice that had been adopted in June, 1931, a complete usable document. The forms were distributed and reviewed by S.A.E. members over a period of time during which the Transportation Division of the Standards Committee was instituted. After trial of the completed report by a number of members, it was finally submitted for approval and adoption by the Society in June, 1934.

The complete report has been prepared by the Society in an $8\frac{1}{2}$ by 11 in. pamphlet form giving the complete classification and definition of cost items, their grouping into three plans, A, B and C as referred to above, and the headings for all the forms that are for use with the Classification. It had been suggested that these forms be printed and carried in stock by the Society for distribution to operators but it was decided not to do so for sufficient fundamental reasons. It will be a simple matter, however, for individual operators to have their own supply of forms printed, using those in the report as models. Copies of this report may be obtained from the Society at a cost of 50 cents per copy for members and at $1 per copy for non-members. Discounts are available on quantity purchases.

Contact!!

THE S.A.E. MEMBER

and

THE EMPLOYER

BRINGING together the unemployed S.A.E. Member and the employer is the principal function of the Placement Service of the Society. The right S.A.E. Member in the right job is an ideal combination that could not fail to bring results, and we need your cooperation in broadcasting to employers the advantages of this Service, which is gratis.

Constant vigilance for available positions will be your contribution to the placement efforts of the Society.

Communicate with your local Section officers or the

PLACEMENT SERVICE

Society of Automotive Engineers

29 West 39th St., New York

Vehicle Design From a Maintenance And Operating Standpoint

By Fred L. Faulkner
Automotive Engineer, Armour & Co.

IT is the opinion among operators of some larger fleets of motor-vehicles that a material reduction in servicing costs could be made, provided vehicle manufacturers cooperated in effecting standards of such items as do not enter into fundamental design, but which affect vitally mechanical servicing.

Sufficient data have been collected to indicate a wide variance of opinion among the fleet operators as to their actual needs. Likewise sufficient data have been accumulated from the manufacturers to indicate a lack of knowledge of the average operator's requirements on their part.

Numerous items requiring constant maintenance have been studied and recommendations outlined with a hope that they will lead ultimately to definite standardization.

It is anticipated that this review of the situation will accomplish at least two things. First, a more unified opinion on the part of the operators as to their actual needs. Second, a better understanding and a more sympathetic attitude to the operator's requirements on the part of the manufacturer.

BY way of explanation, permit me to state that the subject of this paper was assigned to me because of my being chairman of the committee that is studying designs of motor-vehicles from the operator's standpoint.

There has been considerable discussion of this subject at various transportation meetings. It appears to be the consensus of opinion among many operators of motor-vehicles that considerable improvement could be made in their design and construction, which would simplify materially the maintenance and servicing problem and likewise reduce maintenance costs.

The Transportation and Maintenance Committee of the Society considered the subject of sufficient importance to appoint a sub-committee of their group to attempt to bring these comments and criticisms out into general discussion where they could be given the fullest consideration by both the operators and manufacturers' engineers.

I would like to have you accept the comments and criticisms which follow as an expression of many operators with whom the subject has been discussed and their opinions incorporated herein. Several recommendations will follow with which I am not fully in accord; but, on account of the fact that the majority of operators contributing have expressed themselves favorably, which is the determining factor in the tentative standards we are setting up, they have been included.

From the Society standpoint, I would like to have this paper looked upon merely as an outline of weaknesses in design and lack of standardization among the manufacturers of motor-vehicles, on so many seemingly unimportant items about which the vehicle operator is very much concerned.

It is not the intent of this paper, neither is it the purpose of the Committee, to make any recommendations pertaining to fundamental designs. In our preliminary work we are concerned with design only as it affects the ultimate operating cost of the vehicles, primarily from the running maintenance and servicing standpoint. I wish to state further that it is not the intent of our Committee at this time to study designs of the controversial order.

Reflecting for a moment on the progress of the automobile industry, we find that in the space of a very few years the automobile has developed from a pleasure-type vehicle to a business necessity. The motor-truck and motorcoach, which are later developments of the pleasure vehicle, unfortunately have carried with them many elements of design in common with the so-called pleasure-type vehicle. Mechanical failures in the pleasure vehicle are annoying but seldom serious, whereas, with the commercial vehicle, customers disappointed on account of late deliveries due to faulty design or construction, are extremely costly.

As we follow the development of the motor-vehicle and particularly its application to commercial work, we see many adaptations of so-called passenger car designs to motor-trucks, and it is common practice among many manufacturers who produce both passenger-cars and trucks to allow the passenger-car influence to predominate in the motor-truck design.

It has been aptly stated that "engineering designs consist of happy compromises." Unfortunately, too many of these com-

promises made in favor of the passenger-car, when used in truck designs, have proved costly to the operator.

I am not unaware of the necessity of the manufacturer, with a large production of both passenger-cars and trucks, taking advantage of the duplicate use of parts; but it should be done with extreme care if maintenance costs are to be kept within reasonable limits.

To criticise the motor-vehicle manufacturer for the lack of standardization and lack of uniformity of performance, without giving due credit to the fine products we have, would be decidedly unfair. We all appreciate the improvements in motor-vehicle design that have been made, but likewise, we fully realize the opportunity for further economies in operation—if only we could develop a more sympathetic attitude on the part of the manufacturer to the fleet operator's needs.

We recognize that the number of vehicles owned by all of the large fleet operators consists of a very small percentage of the total vehicles manufactured. As a result, the manufacturers as a group will be guided by the demands of the majority. It is therefore necessary for the minority group to do a selling job with the manufacturer, from a standpoint that these changes we want made in design to lower our operating costs will likewise enable them to do a more economical job of servicing, although their vehicles be in the hands of individual owners.

I am not willing to lay all the blame at the manufacturer's door regarding the lack of uniformity of design in the industry, as in my conversation with designing engineers they state that what I may want some other operator does not want, and what the other operator is requesting I do not favor. In other words, there appears to be as much difference of opinion among operators as to what they want or what they need as there is on the part of the manufacturer as to what he builds. We of the Committee recognize our first big job among the operating personnel is to arrive at some agreement among ourselves as to what we need, and then only will we be able to influence the manufacturer to make the desired changes.

With this objective in mind, our Committee set about to study the criticisms of a large number of operators. The results of our studies have confirmed the manufacturer's belief that there is a wide variance of opinion among the operators as to exactly what they want. This is accounted for, however, in the majority of instances, by the vast differences in operating requirements, as our study has covered all types of operation from light-duty to extremely heavy-duty.

In the light duty passenger-car field, a few of the outstanding objections are quoted as follows:

(1) Necessary to remove floor boards to lubricate clutch-release bearings.

(2) No differential drain plug; extremely slow draining.

(3) Filler plug on transmission too close to emergency-brake lever; cannot be removed without difficulty and loss of time.

(4) Necessary to raise hood to fill radiator.

(5) Gasoline filler so arranged that it is impossible to fill tank with any degree of speed.

(6) Very poor rear vision.

(7) Metal guide in front-door split windows very inconvenient for hand signalling.

(8) Oil fill-pipe very difficult to reach, as the opening is higher than the lower edge of open hood, and radiator stay-rods interfere with insertion of oil measure.

(9) Spare-tire bracket mounting poor.

(10) Sheet-metal work too light for commercial use.

(11) Deep-skirted fenders impractical and maintenance cost high to keep in proper condition.

(12) Radiator mounting poor, will not stand hard work.

(13) Steering-column position bad for average-size driver.

(14) Interior trim not suitable for commercial use.

(15) Rear spark-plug difficult to remove.

(16) Bayonet oil-gage located inaccessibly.

(17) Engine too large for economical operation.

(18) Too many bright parts to keep polished.

Heavy Vehicles Present More Problems

Inaccessibility in the heavier type of vehicles, which are harder worked and require more servicing, is even more outstanding.

(19) Radiator and timing-gear case must be removed in order to tighten the main bearings.

(20) Valve tappets very difficult to adjust.

(21) Motor adjustments hard to make because of extremely wide fenders.

(22) Exhaust manifold extends down over valve cover plate making several valves inaccessible.

(23) To replace fan-belt, it is necessary to remove radiator stay-rods, top-hose connections and pull the radiator forward several inches.

(24) Servicing of battery difficult because of very close quarters and concealed negative cable under the cab body.

(25) Headlamp wiring requires unsoldering to remove radiator shell.

(26) Light-duty axle requires dismantling of considerable portion, so that axle can be taken out in order to remove brake-drum for inspection or replacing lining.

(27) Floor and toe-board construction poor, making it very difficult to get at clutch and other parts.

(28) Body bows so installed that when work is done on the body, bows must be cut out and replaced.

(29) Many lubricating fittings are located where they are almost impossible to reach, due to interference of body, etc.

(30) Oil-pump screen cannot be cleaned without removal of lower pan or crankcase.

(31) Water-pump packing in many cases very inaccessible, requiring special tools.

(32) Transmission cover-plate difficult to remove because of practice of passing body stringers directly over transmission.

(33) Clutch inaccessible and should be designed so it could be overhauled without removing motor or rear axle.

(34) Fuses for lighting and horn service very inaccessible in many cases.

(35) Speedometer drive-gears almost impossible to remove on some models.

(36) Too many new tools needed for each new-model vehicle.

(37) Difficult to maintain correct steering geometry on account of lightness in steering members.

(38) Drivers complain of draft in cab due to seat box not being filled in at bottom.

(39) Cab's width inadequate on many models; does not permit of necessary arm freedom.

(40) Difficult to remove spark and throttle-control rods from steering-column.

(41) Poor temperature control in cars; too hot in summer, too cold in winter.

(42) Rapid radiator deterioration due to poor mounting.

(43) Spare-tire carrier mounting poor on many models.

(44) Frame cross-members poorly located for winch mountings.

(45) Power take-off opening wrong side of transmission for some installations.

(46) Insufficient clearance between tire and spring for use of chains.

The above complaints cover many makes and models. The number of cases that have been cited, however, indicate there is room for considerable improvement.

In checking the design, or better, the assembly of many makes and models, we find that many manufacturers are doing a highly commendable job in locating their units in such a manner that they can be serviced quickly and economically.

It is common practice in this country to pull into a service station with a gasoline pump to your left. Is there any logical reason why the oil filler-pipe should not be on the left-hand side of the engine and the gasoline-tank fill-pipe likewise on the left-hand side of the car? Should it be necessary to remove the floor mat and floor boards to check the battery for water level when a small opening in the floor board could be provided for this purpose? Should it be necessary to drop the oil pan, or in some cases to remove the lower crankcase, in order to clean the strainer around the oil pump? Should it be necessary to remove floor boards and clutch housing cover plates in order to lubricate the much-abused clutch throw-out bearings, when a simple lubricant fitting could be provided that would be accessible by lifting the engine hood?

Recognizing that sediment in the cooling system, which deposits in the water jacket, is one of our greatest maintenance evils, would it not appear logical that the block should be provided with either side plates or adequate drains, so that it could be cleaned thoroughly? Many questions of this character are uppermost in the operator's mind, and it is apparent that just as long as conditions, as outlined above, are encountered in current models, the answer is higher maintenance and operating costs. I see no reason why a man working on the grease racks should be required to have more than one wrench to remove the drain and fill plugs on the crankcase, transmission and rear axle for all makes and types of motor-vehicles. Today, a fairly complete set of tools is required.

In order to make a start toward clarifying this situation, a list of questions was prepared and sent to several outstanding operators for an expression as to the items that should be studied and the extent of the study. These data were then prepared in questionnaire form and sent to many outstanding operators for their recommendations as to standardization. These data were compiled and a recommended standard prepared, the results of which will be shown graphically later.

In checking through the returns and comments of various operators, it appears to me that the greatest amount of criticism placed against the manufacturer is not on fundamental designs, but is in reality on lack of coordination of units in making up the final assembly. A preponderance of evidence shows conclusively that there are entirely too many extreme cases of inaccessibility of parts for both minor and major servicing.

Those of us who operate large scattered fleets depend a great deal upon local service for maintaining them. As a result, the purchase of equipment by makes is influenced a great deal by local service. Due to this fact, several makes of vehicles, of the same capacity, will be purchased, which further complicates maintenance standards. The more nearly vehicles in the same capacity ranges are alike, so far as appurtenances and accessories are concerned, the more simplified the servicing becomes.

We do not advocate that all types of vehicles should be so closely standardized that they would lose their identity, but we do feel that if many items of design were standardized among the vehicle manufacturers, there would be a marked improvement in servicing costs.

The scope of our study was so large that it will be impossible to present all the data in connection with it and keep this paper within the limit prescribed by the Society.

I will outline briefly the suggested standards on items on which we have tentatively agreed.

Front Bumpers: Type—channel bar. Height from ground, light trucks and passenger vehicles 17 in., heavy trucks 23 in.

It is obvious that for maximum protection we should have a uniform bumper height for all types of vehicles. However, it is deemed impractical.

Radiator Guards: Many operators recommend and use radiator guards on certain class of their equipment, but it is the consensus of opinion that radiator guards should not be furnished as standard equipment.

Radiator Grilles: Non-essential and contribute to higher maintenance cost without any return for the investment; not recommended.

Chromium Plating: There has been a definite trend on the part of many manufacturers to chrome-plate all trim parts, such as bumpers, head lamps, headlamp tie bar, radiator shell, radiator grille, hood side-doors, cowl beading, windshield frame, etc. It is generally agreed among the operators contributing to this study that bright parts should be held to a minimum of head lamp, door frames and radiator shell.

Radiator: Type—tubular, mounting—rubber; fill opening size 2½ in. Location—inner left-hand side. Provision should be made in sod pan to permit of collecting drainings from radiator without waste, as many operators use permanent type anti-freeze solution.

Engines: The oil filler pipe should be a minimum of 2 in. in diameter, located on the left-hand side of engine. The oil gage should be of stick type and located on left-hand side with sufficient accessibility to assure accurate gaging.

It has been suggested that oil sticks be provided with a flared collar which fits snugly to crankcase when in running position to afford better seal against oil leakage and to prevent foreign matter from working in around the stick.

Crankcase drain opening should be ¾-in. minimum, plug type, with recess for ⅝-in. square head wrench.

The oil-pressure regulating valve should be located on the left-hand side with external adjustment where same is readily accessible. Flywheel timing mark should be visible from the front right-hand side of engine.

Exhaust-manifold flanges were given very serious consideration but due to variation in design we are offering no recommendation as regards their standards. It is hoped, however, that designing engineers will give this matter serious consideration from the standpoint of stud diameter and available gasket area.

Clutch-Housing should be provided with external fitting that is accessible from under right side engine hood for lubricating clutch throw-out bearings.

Transmission: Filler opening. Location—left side. Transmission case at lubricant level. Size—minimum ¾-in. diameter. Type—elbow. Drain opening, size ¾-in. minimum, plug recessed for ⅝-in. square-head wrench.

Rear Axle: Filler opening should be minimum ¾-in. diameter located center of cover at lubricant level. Plug should be recessed for ⅝-in. square head wrench. Drain opening located at bottom center. ¾-in. diameter minimum plug—

recessed for ⅝-in. square-head wrench.

Cabs: Two-man type. Mounting—three-point suspension. Ventilation: top of cowl in center, minimum. Dimensions: inside height 50 in. Inside width: minimum 50 in. Inside back to dash 36 in. Height: floor to top seat cushion, 13 in. Height: floor to steering wheel, 22 in. Door width: minimum, 31 in. Door hinge at front.

Instrument Board: It is recommended that all control devices be removed from the steering wheel and the instrument board be equipped with the following—speedometer, oil pressure gage, head indicator, ammeter, gasoline gage, choke control, throttle control and light switch. It is further recommended that high and low-beam lights should be controlled by a foot switch, located at left of clutch pedal.

Gasoline Tank: Two recommendations are necessary for location of tank, due to wide application of passenger-type chassis for commercial work. Light-truck and passenger-vehicle tanks to be located at rear of frame. Large-truck tanks to be located under cab seat. Fill openings to be of elbow type, minimum openings 2 in. diameter. Location—left side outside cab. Stand pipe to be baffled. Tank cap to be of bayonet type. Tank drain ½ in. plug.

Lubrication Fittings should be of the zerk snap-on type.

Air Cleaners: An adequate air cleaner of the oil-bath type should be standard equipment on all types of vehicles.

Oil Filters: An adequate oil filter of the cleanable type should be standard equipment on all types of vehicles. Location—left side engine, accessibly mounted for cleaning.

Horn: A horn of electric type, located under engine hood, is recommended. We are not prepared at this time to make any recommendations as to rating of a horn for general commercial use.

Windshield Glass: Shatter-proof safety glass should be standard for all cab windshields.

Windshield Wipers: All cabs should be equipped with two wipers. Method of drive—optional. Vacuum preferred.

Rear-View Mirrors: Type of bracket—tubular. Length adjustable with minimum of 12 in.

Front Fenders: The conventional type of front fender appears to be generally satisfactory. However, many operators are requesting that a coach type fender be made optional on the part of the manufacturer.

Wheelbase and Frame Length: This subject has been reviewed again among the operators contributing to this discussion and it has been generally agreed that the present S.A.E./C.A. dimensions are satisfactory. It is hoped, however, that a larger number of manufacturers will incorporate the present C.A. standard dimensions into their present line of motor-vehicles.

Battery Location: There is considerable variance of opinion as to the proper place to locate the battery. If under the floor board, it should be on the right-hand side; if external, left side of frame. Wherever mounted, to be accessible and as far from muffler pipe as possible.

Spare-Tire Mounting: The problem of satisfactorily mounting the spare tire has perplexed the industry for years. We are unable, at this time, to make any definite recommendations. It is pretty generally agreed, however, that on light-duty equipment we could carry the spare tire in a fender well, left side of the vehicle, and on the heavier type truck resort to the under-side type of carrier to the rear of rear axle. In fender-well type mounting, more attention should be given to proper support of tire to prevent excessive maintenance at this point. On the under-side type carrier, the weight of the tire must be taken into consideration as most road changes are accomplished by one man. The hinged type appears to more nearly meet this requirement.

Tail-Lamp and License-Bracket Mounting: Combination tail-lamp and rear license-bracket should be located external to left-side frame rail, 6 in. from end. It is hoped the designing engineers will give some consideration to the length of the present license plates in mounting this bracket to prevent interference with the plates when removing spare tire.

Many States require both front and rear license-plates and it has been noted that some manufacturers make no provision for carrying front plates. Suitable brackets should be provided to carry front plates either on front bumper or headlamps tie bar.

Location of Tool Box: Provision should be made for tool compartment under cab seat at the right end of tank.

Front Axle Width and Turning Radius: Many operators are demanding wider front axles. We have no recommendations to make regarding what this should be. However, I am mentioning these items as ones that should be given further consideration. The major difficulty seems to be brought about in oversizing tires beyond the manufacturer's standard.

To obtain a measure on how far out of line the industry was in their present day offerings as compared to what the operators have requested, a questionnaire was prepared and sent to the majority of the leading manufacturers. This questionnaire was limited to the service items and those pertaining to the safety of the vehicle. Unfortunately we did not get in sufficient returns to make the data fully representative; it is apparent, however, there is considerable room for improvement.

It is hoped this review of the situation will accomplish at least two things: First, a more unified opinion on the part of the operator as to his actual needs and, second, a better understanding and a more sympathetic attitude on the part of the manufacturers to the operator's requirements.

Summary of Questionnaires to Truck Manufacturers

Number of Truck Manufacturers receiving questionnaire	25
Number of Truck Manufacturers reporting	11
Number of 1934 models represented	30

Editor's Note—Recommended standard size or type desirable was in each case determined from votes cast by 40 fleet operators and automotive engineers engaged in fleet operation throughout the United States[1]. Figures after "Recommended Standard" refer to number of votes favoring the recommendation.

Oil-Fill Location

Location—	Left	Right	Top
Number Models Using	29	0	1

Recommended Standard—Left (37)

Oil Gage

Type—	Bayonet	Float	Other types represented
Number Models Using	30	0	0

Recommended Standard—Bayonet (38)

Of the 30 reported models using the bayonet gage there are 12 different types of oil level markings used.

Oil-Fill Size

Diameter—In.	1 3/16	1 1/4	1 1/2	3	2	1 5/8
Number Models Using	2	4	19	3	1	1

Recommended Standard—2 in., minimum (38)

[1] A more detailed summary of these is given in tabulated "Results of Questionnaire to Fleet Operators" printed with this paper.

Crankcase Drain

Diameter opening—In.	3/4	7/8	11/16	7/16	1/2
Number Models Using	13	6	6	2	2

Recommended Standard—3/4 in. plug, Recessed for 5/8 in. square head wrench (37)

Gasoline Tank Location

Location—	Under Cowl	Driver's Seat	Frame Rear	Frame Side
Number Models Using	0	27	1	2

Recommended Standard—Light trucks—Frame Rear
Heavy trucks—Under Driver's Seat (35)

Gasoline Tank Fill Opening

Location—	Inside Cab	Outside Cab
Number Models Using	23	7

Recommended Standard—Outside Cab, with baffled standpipe (32)

Gasoline Tank Fill

Diameter—In. Fill Opening	1 1/2	1 3/4	2	2 1/8	2 1/4	1 27/32
Number Models Using	4	13	1	2	8	2

Recommended Standard—2 in. minimum (36)

Gasoline Tank Cap

Type	Threaded	Bayonet	Spring
Number Models Using	17	10	3

Recommended Standard—Bayonet type (36)

Gasoline Tank Drain

Diameter—In.	1/8	1/4	1/2	No Reply
Number Models Using	7	9	3	11

Recommended Standard—1/2 in. pipe plug (31)

Radiator Fill

Type—Fill Cap	Threaded	Bayonet	Other type
Number Models Using	6	23	1

Recommended Standard—Bayonet type, external of cab

Diameter—In. Fill Opening	1 1/2	1 3/4	1 17/32	1 3/4	1 7/8	2	2 1/4	2 1/2
Number Models Using	2	3	3	4	1	14	1	2

Recommended Standard—2 1/2 in. (24)

Radiator Drain

Type of Drain	Needle Seat	Pipe Plug	Petcock
Number Models Using	1	2	27

Recommended Standard—Type —Needle Seat
Size —3/8 in. I. P. T.
Location—Inner Left (34)

Radiator Mounting

Type	Rubber	Spring	Other
Number Models Using	6	20	4

Recommended Standard—Rubber (27)

Transmission

Fill Opening Location	Full Lub. Level, Left	Full Lub. Level, Right	Top of Gear Case	Other
Number Models Using	3	26	1	0

Recommended Standard—Full Lubricant Level, Left Side (37)

Fill opening Diameter—In.	3/8	1/2	3/4	1
Number Models Using	3	3	21	3

Recommended Standard—Min. 3/4 in. Diameter —Elbow (35)

Drain Opening Diameter	3/8	1/2	3/4	1
Number Models Using	3	3	21	3

Recommended Standard—Min. 3/4 in. Diameter (35)

Drain-Plug Size

Recommended size wrench to remove fill-and-drain plugs —5/8 in. Square-Head Wrench.

Rear Axle Fill Opening

Diameter—In. Fill Opening	1/2	5/8	3/4	1	1 1/2	1 1/4	N.R.
Number Models Using	2	1	15	5	1	2	4

Recommended Standard—3/4 in. Diameter (36)

Diameter Drain Opening	3/8	1/2	3/4	1	1 1/2	N.R.
Number Models Using	5	5	12	3	1	4

Recommended Standard—3/4 in. Diameter (36)

Recommended standard wrench to remove rear axle Fill and Drain Plugs—5/8 in. Square-Head Wrench.

Two-Man Cab

Dimensions—In.	Variation in 30 Models	Recommended Standard
Inside Height	46 to 57	50 in.
Inside Width	49 1/4 to 68 1/2	50 in.
Inside Back to Dash	28 3/8 to 53	36 in.
Floor to Top Seat Cushion	12 1/2 to 17	13 in.
Floor to Steering Wheel	21 1/2 to 26	22 in.
Door Width	26 to 33 1/2	31 in.
Door Hinged, Front Rear		Front

Safety Glass

Used In—	Windshield Only	All Glass in Cab
Number Models Using	5	0

Recommended Standard—Safety-glass windshield in all cabs; other glass optional.

Lubrication Fittings

Type—	Alemite	Zerk	Snap-On Zerk
Number Models Using	4	17	19

Recommended Standard—Snap-On Zerk (26)

Windshield Wipers

Type	Vacuum	Electric	Mech. Power
Number Models Using	26	4	0

Recommended Standard—2 Wipers, Vacuum or Electric (34)

Results of Questionnaires to Fleet Operators (40 Operators Voting)

	Voted Standard		Other Recommendations		Optional	No Reply
ENGINES						
Filler and Gage Location	Left	37	Right	0		3
Filler and Gage Size	2 in. Minimum	39		0		1
Gage Type	Bayonet	38		0		2
Drain Type	Plug	40		0		0
Drain Size	¾ in.	38	Larger	2		0
RADIATOR						
Type	Tubular	37		1		2
Mounting	Rubber	28	Spring	7	2	3
Fill-Opening Size	2½ in.	25	2 in.	8	2	5
Drain-Opening Size	⅜ in. IPT	35	Larger	2		3
Drain-Opening Type	Needle Seat	35		2		3
Drain-Opening Location	Inner Left	40		0		0
TRANSMISSION						
Filler Opening Location	Left Liquid Level	38		0		2
Filler Opening Size	¾ in.	36	Larger	4		
Filler Opening Type	Elbow	30	Side Case	8		2
Drain Opening—Size	¾ in. Plug	36	Larger	4		
Size Wrench to Remove Plug	⅝ in. Square Head	36				4
REAR AXLE						
Filler Size	¾ in.	37	Larger	2		1
Filler Location	Rear Cover Liquid Level	38		0		2
Drain Size	¾ in.	37		2		1
GASOLINE TANK						
Location	Small—Rear Large—Seat	35	Side of Frame	2		3
Fill Opening	Elbow	36		0		3
Cap Type	Bayonet	37		0		3
Fill—Size	Minimum, 2 in.	37	Smaller	0		3
Fill—Location	Left—Outside Cab	34	Inside Cab	0		6
Drain Plug	½ in. Pipe Plug	33	Other Type	1		6
LUBRICATION FITTINGS	Zerk	32	Alemite	12		
AIR CLEANERS STANDARD?	Yes	39	No	0		1
Type	Oil Bath	39		0		1
OIL FILTERS STANDARD?	Yes	39	No	0		1
Type	Cleanable	12	Cartridge	3	2	23
Location	Left Side	28	In crankcase	1		11
REAR VIEW MIRRORS						
Type Bracket	Tubular	37	Other types	1		2
Length	Adjustable	37	Solid	1		2
	Minimum Length 5 in. or less	6	Minimum Length 12 in. or less	11	Minimum Length more than 12 in. (3)	20
WINDSHIELD WIPERS						
Number	Two	35	One	3		2
Type	Vacuum	34	Electric	3	Mechanical (1)	2
HORN						
Type	Electric	33		0		7
Location	Under Hood	29	Outside	3	1	7
RADIATOR GRILLES	No	27	Yes	3	2	8
RADIATOR GUARDS	No	28	Yes	3	2	7
CHROME PLATING	No	31	Yes	2	6	1
BATTERY						
Location	Right—Under Floor Boards	28	Other Location	12		0

Results of Questionnaires to Fleet Operators (40 Operators Voting)

	Voted Standard		Other Recommendations		Optional	No Reply
SPARE TIRE MOUNTING						
Location Spare	Small—Fender Well Large—Frame, Rear	8		0	3	29
TAIL LIGHT AND LICENSE	Left—Under Frame 6 in. Ahead Rear Cross Member	35	Other Location	2		3
CABS—TWO MAN						
Mounting	3 Point	34	4 Point	2		4
Ventilation	Center Top Cowl	37	Additional	24		3
Dimensions:						
Inside Height	50 in.	36		0		4
Inside Width	50 in.	36		0		4
Inside Back to Dash	36 in.	32	More than 36 in.	4		4
Floor to top seat cushion	13 in.	36		0		4
Floor to steering wheel	22 in.	34	More than 22 in.	2		4
Door Width—Minimum	31 in.	34	Less than 31 in.	1		5
Door Hinged	Front	36	Rear	0		4
INSTRUMENTS ON DASH						
Speedometer						
Oil Gage						
Heat Indicator						
Ammeter						
Gasoline Gage						
Light Switch						
Choke Control						
Light Control with Foot Controlled Dimmer	All Standard	38		0		2
FRONT BUMPERS						
Type	Channel Bar	28	Other	3	9	
Height from Ground	Light—17 in. Heavy—23 in.	31	Same Height	9	0	
SAFETY GLASS						
Windshield	Yes	26	No	0	14	
All Glass in Cab	Yes	4	No	0	36	
SHOULD WE INTEREST OURSELVES IN:						
1. Location of Oil Pressure Regulating Valve?	Yes	20	No	15	5	
2. Flywheel Timing-Mark Opening in Bell Housing?	Yes	30	No	3	7	
3. Exhaust-Manifold Flanges?	Yes	16	No	12	12	
4. Lubrication Fittings for Clutch Throw-Out Bearings?	Yes	23	No	5	12	
5. Location of Tool Box?	Yes	17	No	17	6	
6. Front-Axle Width and Turning Radius	Yes	18	No	9	13	
IS PRESENT S. A. E. HEAD-LAMP MOUNTING SATISFACTORY?	Yes	34	No	1	5	
WHEELBASE AND FRAME LENGTH:	S. A. E.—C. A. Dimensions satisfactory by almost unanimous vote.					

Discussion

Sees Valuable Results From Committee's Work

—Clinton Brettell
Superintendent of Garages, R. H. Macy & Co., Inc.

I REGRET my inability to attend the meeting, and to personally participate in the discussion of this very timely and thorough paper. I am glad, however, of the opportunity to present a few thoughts on the subject.

First of all, I think Mr. Faulkner is to be highly com mended on his courage in tackling such a controversial problem, and for the businesslike manner in which he has accumulated the basic facts, from a typical cross-section of users and builders of commercial vehicles.

In reviewing his two summaries it is quite apparent that although there is considerable variance in the opinions of operators on the several items the practices and standards of manufacturers are even more divergent.

The work of the sub-committee should result in the crystallization of "operators' needs" into a condition of much greater uniformity, and then the "sale" of this idea to the manufacturers.

This latter result can undoubtedly be materially accelerated if we, as individual users, let it be known to the manufacturers with whom we have contact, that it is our individual and combined wish that definite steps be taken by them to accomplish these results.

And, of course, not forgetting the obligation on our part, to study this matter with a view to securing a more intimate knowledge of our requirements and methods of simplifying and standardizing same.

The ultimate goal of such a movement as that of this committee should be the promotion of a much closer co-ordination between manufacturer and user—and, to this end, the employment of field engineers, to serve as a liaison unit between the two, would be a long step in the right direction.

I look with great hope for a splendid achievement of the committee under the able leadership of its Chairman, the author of this paper.

Offers More Points Worth Considering

—Harry O. Mathews
Superintendent of Motor Equipment, Illinois Bell Telephone Co.

I HAD the pleasure of contributing a small part of the information referred to as having been secured from the operator, and in my opinion, the most important points in design affecting maintenance costs have been covered.

One point which has recently come to my attention is the very inaccessible location of the filler plugs on shock absorbers on certain models. On the models in question it would have been quite easy to locate these plugs in a position more accessible to the service man.

Another problem in design was forcibly called to my attention when we found it necessary to change the location of the exhaust pipe on a large truck to permit the installation of a power take-off on the side of the transmission. This manufacturer agreed to make a change in design to correct this point.

As we service most of our vehicles in Chicago ourselves, it is constantly brought to my attention that too much time is required to grease and oil the cars. Although the time element is a direct reflection of the cost, a very important fact is that the service man is inclined to pass up the inaccessible points. Many clutch throw-out bearings have been replaced on that account. While this is only one item, I believe that it is representative of maintenance troubles occurring from faulty design.

Many of the other points covered are quite important to us and it seems to me that if the design were changed to meet the requirements of the fleet operator it would also meet the requirements of the private owner who does his own service work, or the service station which does it for him.

Among the most important points which I think should be considered are:

(1) Location and design of gasoline filler-pipe.
(2) Location of battery.
(3) Location of radiator drain.
(4) Location of tail lamp and license bracket.
(5) Air cleaners and oil filters.

Factors Affecting Accidents

THE National Safety Council reports that in 1933, 7 per cent of the motor vehicles involved in fatal accidents were reported defective in some particular. Of these, 2½ per cent had defective brakes, 2 per cent improper lights, and ½ per cent had defective steering mechanism. This does not show the full extent of mechanical failures due to some vehicles being damaged too badly to identify defects and the unwillingness of live drivers or the inability of dead ones to admit them.

Harold G. Hoffman, Commissioner of Motor Vehicles for the State of New Jersey, states that their records over the past few years indicate that 6 per cent of New Jersey accidents are caused by defective mechanism of cars or faulty road construction.

The Committee on Motor Vehicle Maintenance of the National Conference on Street and Highway Safety estimates that defects contribute in 15 per cent of all accidents.

In commercial fleets, the percentage of mechanical defects contributing to accidents is low. In the Oklahoma Gas and Electric Company for instance, H. R. Grigsby, Superintendent of Transportation, informs us that in the past 2½ years, there have been only three accidents, in 9½ million miles of operation that could be attributed to mechanical defects. Mr. Grigsby's fleet had an accident frequency during the period of 1.215 compared with the national average of 2.42 accidents per 100,000 miles in public utility operations, taking first honors in the National Safety Council Fleet Contest. . . .

Frequent and careful inspection is apparently one of the cheapest methods of minimizing accidents. Vehicles must be inspected periodically to insure of their being in fit condition for the service to which they are assigned.

—Excerpts from a 1934 Semi-Annual Meeting paper on "Accident Control in Fleet Operation" by J. M. Orr, Equitable Auto Co.

Discussion of S. W. Sparrow's Paper on Main and Connecting-Rod Bearings

MR. SPARROW stated in his paper that copper-lead mixtures are superseding babbitt to some extent for main and connecting-rod bearings, and he discussed these primarily from the standpoint of their performance in comparison with that of babbitt.

The paper was published in the July, 1934, S. A. E. JOURNAL, pages 229 to 237 inclusive.

Comments are made in the following discussion of the paper on the various subjects treated therein.

Categorical Comments on Bearing Materials

—Robert G. N. Evans
Bunting Brass & Bronze Co.

MR. SPARROW deserves praise for his work and the way in which he brings attention to existing conditions.

Babbitt

At the beginning of his paper, Mr. Sparrow places his finger on the father of bearing failure—inadequate or ruptured oil film—and refers to babbitt as being the accepted material, at least up to now, believing this metal to be kinder to the shaft when the oil film is broken. There are many babbitts, which differ in both chemical and physical properties, that have been developed in the past, all in an attempt to increase their ability to carry loads at elevated temperatures. Additional elements have stepped up hardness and delayed softening, somewhat. Hardness in the babbitts has been raised to 20-25 Brinell at room temperature; but, at temperatures approximating 300 deg. fahr., at which we operate, hardness will show 6-12 Brinell. This hardness value accounts for the results mentioned by Mr. Sparrow. Our experiments would indicate that the higher-copper babbitts would have the most endurance, but all are hopelessly inadequate to support the load and stand up under temperatures that we know exist. See Fig. 1.

Cracking.—I would refer to the failure as fatigue caused by softening and flexing. I believe we have few cases of melting, excepting desert work under heavy loads.

Bond.—I agree with Mr. Sparrow that poor *bond* is a fault of manufacturing procedure rather than a lack of knowledge.

Deflection of Bearing Shell.—There can be no question that many good bearings are ruined by poor backing. Here let me say that I think the engineers were sold something when they adopted thin-wall bearings. I shall qualify this statement later. Mr. Sparrow's findings in regard to crush on bearing halves at the parting line agree with ours. Flexing of a bearing, if mounted in intimate contact over its entire surface, is a function of inadequate strength of the support.

Temperature.—Many changes of chemical properties in babbitts have failed to increase the value of heat conductivity materially. The coefficient of thermal conductivity of babbitt is very low; therefore, heat generated is retained at the rubbing surface. Heat is the greatest factor in the destruction of oil viscosity. With a drop in viscosity more volume and pressure are required to keep the lubricant present.

Plastic bronze has a high thermal conductivity, which is certainly one of the requirements of a good bearing metal and when it is placed in intimate contact with its backing, its rapid transmission of heat is contributory to long life. We made comparisons between babbitt and plastic bronze with a thermocouple sunk in the bearing metal close to the rubbing surface and the bronze showed a drop of 32 deg. fahr. under that of babbitt, this being comparable with the increase in crankcase wall-temperature. Tests were conducted with wide-open throttle at 3800 r.p.m. and full loads; the duration was 105 hr. of continuous operation.

Other Causes of Breaking.—As Mr. Sparrow states, all failure of the babbitt-lined bearing cannot be laid to faulty bond or flexing of the shell, and he offers his experience with reduction of bearing surface as proof. Here, he accomplished two things by removing the load from the edges of the fused surface. He prevented metal work or flexing of the lining at the outer edges caused by shaft deflection, and he lessened the chance of the oil wedge parallel with the bearing axis. We have found that larger diameters and shorter lengths, within limits, produce better oil-film coverage and fewer bearing failures. Shortening the length of the rubbing surface was a step in the right direction.

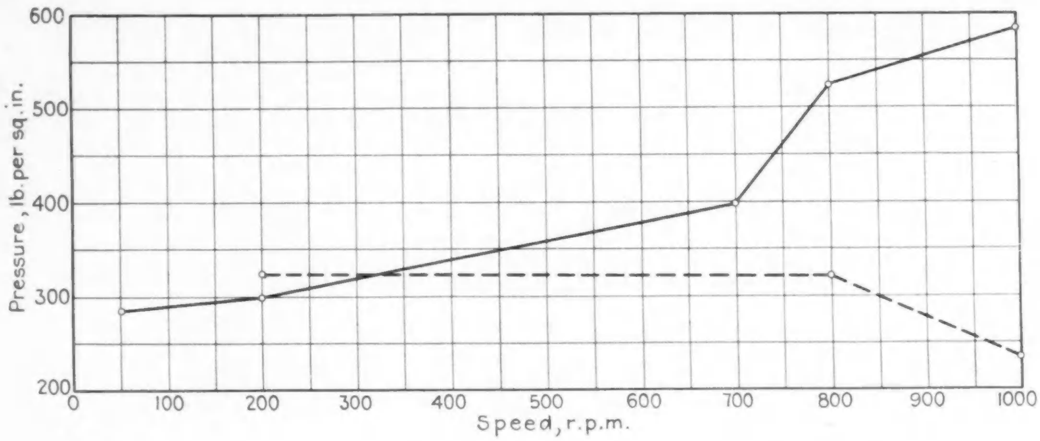


Fig. 1—Comparative Curves of the Performance of Two Bearing Materials

This is an adaptation of results obtained by Professor Moffitt, of Ohio State University. The solid-line curve is for Bunting Brass & Bronze Co. bearing material No. 72; the dash-line curve, for S.A.E. No. 11 babbitt. The shaft was cold rolled. The temperature of the lubricating grease was 180 deg. on all bearings.

Thickness of Babbitt.—The ability of bearing metals to carry shock loads increases as the wall thickness decreases and, if 0.006 in. of babbitt was used, unfused areas would evidence themselves in manufacturing procedure.

Undercutting to Extend Life.—This is true without question. Working of the babbitt structure, except under a long period of maximum stress, does not extend into the undercut section. The *P-V* or *ZN/P*-relationships constitute only one of the factors in bearing calculation.

Copper-Lead Bearings.—Copper-lead or plastic bronzes will eventually be the bearing metal used. It has, as Mr. Sparrow says, a higher melting point and a lower coefficient of friction at elevated temperatures, see Fig. 2, and a greater resistance to deformation than do the babbitts.

Life.—Long life is a result of using correct bearing material, shaft finish, hardness and oil film.

Overheating.—Mr. Sparrow's summing up of this question is directly in line with our experience. In one test which we made on a one-mile dirt-track, for a test with plastic-bronze main-bearings, we were limited to 1300 miles on babbitt-lined rods. These were replaced with plastic-bronze liners leaving the crankpins as the babbitt left them, cutting down the oil inlet, allowing for reduction in bearing size under temperature and increasing running clearance. We were then able to complete a 76,000-mile test without losing a rod; the bad track condition stopped the run.

Disintegration.—Changes in the elevation of the bearing surfaces, from any cause, crack the oil film. One of these causes is bi-metallic differences in expansion.

Unsuitable Oil.—This washing away, as described by Mr. Sparrow, does exist in mixtures of copper and lead with lubricants containing fats or acids; but, so far it has evidenced itself only slightly with our plastic-bronze alloy, due, we believe, to the fact that an additional alloying element is present. Certainly, the use of copper-lead bearings is restricted, if not entirely forbidden, if they are vulnerable to the action of some oils placed on the market for consumers' use. It might be well at this time to lodge a protest against the use of so-called compounded oil that leaves the engine ruined after the break-in period. The way most engines are built today, an S.A.E.-30 oil will allow periods of 60 m.p.h. without harmful effects. Why use a destructive break-in oil?

Bearing Shells.—A bearing is no better than its backing, and its wall thickness should not be determined by allowable space left between shaft and housing.

Sequels to Thin-Wall Bearings

I stated earlier in this discussion that engineers had been sold something when they adopted thin-wall bearings. Let me qualify this by saying what they really bought was the chance to increase shaft diameter without increasing the size of hole or case at the expense of closer machine limits in the case—questionably close contact between bearing back and

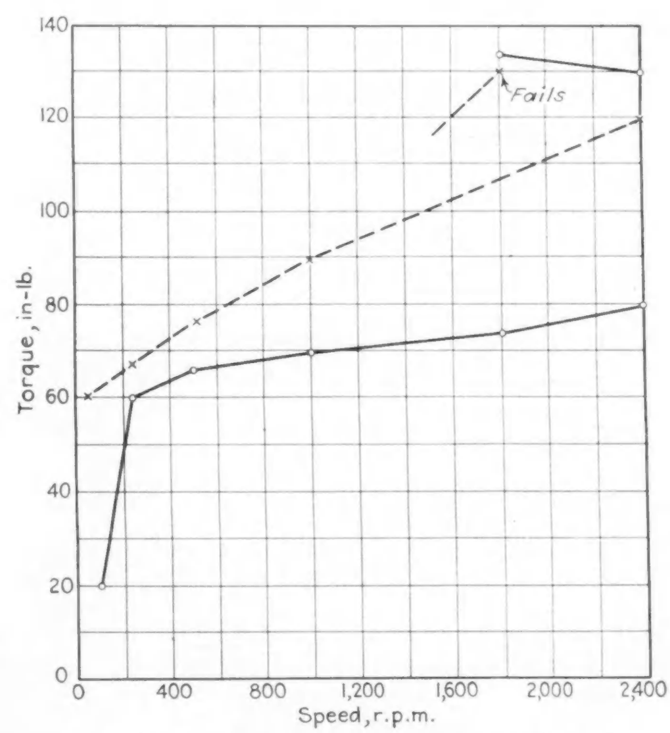


Fig. 2—Comparative Curves for Bronze and for Babbitt

This is an adaptation of results obtained by Professor Moffitt, of Ohio State University. The solid-line curve is for bronze; the dash-line curve, for babbitt. The load was 225 lb.

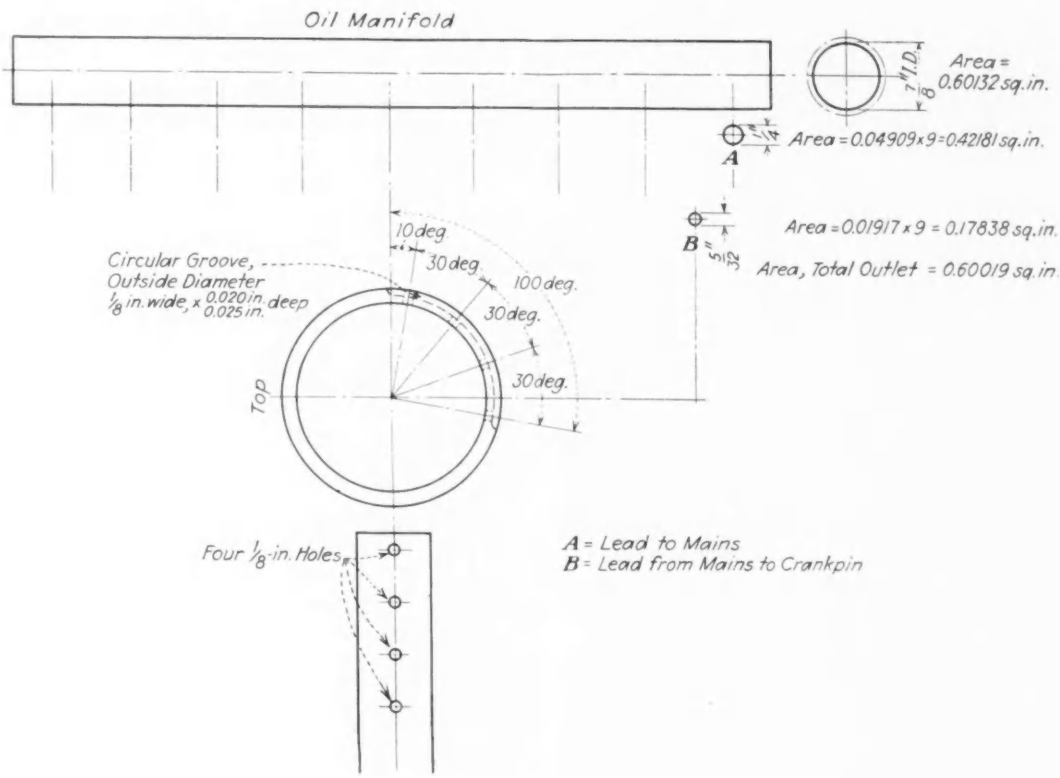


Fig. 3—Method of Supplying More Oil to a Crankpin

block without undue strain on studs or cap screws—block squirm and strain on the bearing structure excited by shaft deflection limited to 0.016-in. thickness of babbitt instead of the entire thickness of the wall had it been made with a bronze backing, and also eliminating the line ream after assembly which corrects all errors which creep in. Some manufacturers have returned to the final line-reaming, but use the precision type of bearing, removing stock in the bore. This fully interchangeable bearing sounds well in advertising, but certainly does not guarantee the parallel rubbing surface which is needed for preserving the oil film. Crush can be taken on a bronze-backed or all-bronze bearing without permanent set of the bearing or strain on studs or case. Heat dissipation was mentioned under the heading of *Temperature*.

Oil Demand Rises with Speed

As Mr. Sparrow points out, oil demand is greater with increased speed in revolutions per minute. To control this, the area of all oil outlets should not exceed but should rather be less than the area of the oil manifolds so that not only will a pressure be maintained but one bearing with greater running clearance will not starve the other.

In Fig. 3, no attempt has been made to specify exact size but simply to illustrate the thought and method of supplying more oil to the crankpin. This same method of grooving and drilled hole can be used to supply more oil to the piston-pin bearing, but the groove on the back should cross the parting line on the right-hand side. With this method of grooving the outside diameter of the bearing and drilling registering holes, we increase the number of times the holes in the shaft and the bearing register—that is, can be carried entirely around—***without breaking up the oil film*** or reducing the area of the rubbing surface, and making leakages at the ends of less concern. A similar method was suggested by Alex Taub at the 1932 Annual Meeting in Detroit.

Engine temperatures must be given thought if builders of cars are going to tuck them away, as Mr. Sparrow states. Perhaps water-jacket lengths will increase or some form of forced draft will be employed to carry off heat radiated to the block and crankcase.

Dirt.—In the engine, dirt is a factor. We have fed sand and dust to engines on our block tests to approximate road conditions and naturally find the bearings scratched at points of oil entry. Most bearings have a low-pressure side and allow oil spill. Where the motion is oscillating and has little running clearance, grooves out through the ends on the low-pressure side assist in dispelling dirt. Dead ends should be avoided. An oil filter of any kind helps as long as it is kept clean; it also has a tendency to reduce oil temperature.

Mr. Sparrow's paper shows a practical knowledge of his subject and, if heeded, will be of material benefit to automotive and lubrication engineers.

The Airplane Landing-Gear Shock-Absorbing System

By C. V. Johnson
Bendix Products Corp.

THIS paper points out that the shock-absorbing system of the main landing-gear of an airplane must function under the impact of landing and while taxiing on landing fields. The present requirements for impact landing are outlined, and a typical analysis is made to check up a proposed system for a given airplane.

The effects of geometrical arrangement of gear, of tire size and of tire rebound are considered. Laboratory methods of testing systems to determine whether the requirements have been met are discussed.

The characteristics desired of the shock-absorber system for good taxiing are enumerated, and the effects of various types and arrangements of absorber units on performance are investigated.

In conclusion, the paper presents a discussion of the tail-wheel shock-absorber system in which it is brought out that the same criteria that are applied to the main gear apply here, but that the relative importance of various factors is not the same as in the main gear.

THE airplane landing-gear, including both main and tail-wheel gear, has been under indictment on two major counts in recent years. It has been accused of causing as much as 20 per cent of the drag of the airplane while in flight, without serving a single useful purpose in the flight condition. That this was true is indicated by the numerous retractable gears that have appeared recently. The second count is that 70 per cent of the structural failures of the airplane causing accidents are in the landing gear. There is some question about the seriousness of this last accusation because, in the event of a landing hard enough to cause some structural failure, and these will occur, the failure should be in the landing gear. The percentage of total structural failures which are in the landing gear should therefore be high.

The real criterion of improvement will be in the reduction of structural failures per hour of service. The factor having the greatest influence in eliminating landing-gear failures is an adequate shock-absorbing system; that is, one having the capacity of absorbing large quantities of energy and at the same time keeping the developed loads within a safe figure. It is the purpose of this paper to discuss the shock-absorbing system of both the main gear and the tail-wheel gear under both the landing impact and the taxiing loads to which they are subjected.

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

The impact landing-condition for the main landing-gear will serve as a starting point, since it is the one usually given the greatest weight in design consideration. As in all engineering problems, it is necessary to set up a requirement which the design must meet. Logically, this would be set up as the capacity to absorb the energy, without exceeding certain established loads, of an airplane approaching the ground with a velocity having a certain vertical component. Knowing the weight of the airplane and the vertical velocity component, it is easy to determine the kinetic energy to be absorbed at the moment of ground contact. This energy is the exact equivalent of this same weight falling freely through a distance which will give it, at the moment of contact with the ground, a velocity equal to the vertical component of the airplane's velocity as it approaches the ground for the landing. From this relation was established the "drop-test requirements" which have long been used as the criteria for the design of the shock-absorber system.

Stated in simplified form, this requirement is that the landing gear shall be capable of absorbing the energy developed by allowing the gear carrying the weight of the airplane to fall freely through a given distance without developing loads in the gear in excess of given factors. The statement of the requirements as established by different countries and different agencies in the same countries vary in details. It is not the purpose of this paper to discuss these differences or the question of whether they represent operating conditions, although this would be a splendid field for turning on the searchlight of recent experience and theory. For our purpose it will be sufficient to give the requirements as established for civil aircraft by the Aeronautics Branch of the Department of Commerce. The requirements established by the Army and Navy are the same type, but are more severe, due to the greater hazards of military operations, and particularly to the hazards of landing on aircraft-carrier decks with their arresting gears.

The proposed amendment to Bulletin 7-A to be effective July 1, 1934, makes the following requirement:

"Section 39.—Shock-Absorption Requirements.—(A) The height of free drop, in inches, shall be 0.36 times the calculated stalling speed in miles per hour, except that it need not be greater than 18 in. The height of free drop is measured from the bottom of the tires to the ground, with the landing gear extended to its extreme unloaded position.

"*(B)* The shock-absorbing system, including that for the tail, shall be capable of absorbing the energy generated during the entire vertical travel of the airplane when the airplane is dropped from the required height of free drop, without subjecting any structural member to a load greater than its maximum design-load. This applies to both the level-landing and three-point-landing conditions. The effects of support from the wings and the shock-absorbing capacity of the structure shall be neglected in determining the energy to be absorbed by the shock-absorbing system."

Note that this requirement specifically excludes considering the aircraft structure as a part of the shock-absorbing system. This simplifies our problem, since the allowable shock-absorbing capacity of a structure is indeterminate, particularly when fatigue is considered. It means that, in design considerations, the landing-gear shock-absorber system consists of the tires and the shock struts.

The problem of determining the required energy-absorption capacity of a system consisting of a tire and strut to meet the established drop-test requirement lends itself to a simple mathematical analysis which will be followed through for a typical case. Fig. 1 gives a line diagram of one type of gear. The axis of rotation is at right angles with the longitudinal axis of the airplane; hence, only one view is necessary for our purpose. Note that the tire is a standard 36 x 8 inflated to a pressure of 60 lb. per sq. in. and that the total load of the airplane is 8000 lb. The stroke of the strut is 6 in. and it is vertically over the wheel in the front view. The size and type of strut has no bearing on our analysis of requirements. From the Department of Commerce Bulletin 7-A, we determine that the system must have capacity enough to absorb the energy of a free drop of 18 in. in either the three-point-landing or level-landing condition without building up factors in excess of 5.33.

Fig. 1 shows conditions as they exist for both types of landings. Under the three-point landing set-up, the actual load on each landing-gear wheel is 3600 lb. This is due to part of the load being carried by the tail wheel. The total movement of the center of gravity of the airplane due to the stroke of the oleo is 6 in. This is actually measured at the wheel. The possible maximum deflection of the standard 36 x 8 tire is 6 in. and, at this deflection, it stores 40,000 in.-lb. of energy as determined by test. The actual mass-travel, assuming maximum tire deflection and strut deflection occur together for an 18-in. free-drop, is 18 in. (drop) + 6 in. (strut stroke-movement) + 6 in. (tire deflection) = 30 in. The total energy to be absorbed by each wheel and strut is 30 in. × 3600 lb. = 108,000 in-lb. If the tire absorbs 40,000 in-lb., the strut must absorb 68,000 in-lb. The travel of the center of gravity due to the oleo stroke is 6 in. The average resistance for this full stroke must therefore be 11,333 lb. If the average resistance is 80 per cent of the maximum —which would represent a good strut—the maximum resistance is 14,200 lb. Based on the load of 3600 lb., this represents a load factor of 3.94. This is well within the permitted load of 5.33.

For the level-landing condition, the requirement specifies the introduction of a drag component which brings the reaction through the axle and the center of gravity of the airplane. This condition is represented by rotating the airplane about the axle until the center of gravity is vertically above the axle and the full weight of 8000 lb. is on the wheels. The shock-struts' new position is such that the full stroke of 6 in. only gives a vertical movement of the center of gravity of 4 in., as in Fig. 1. The tire deflection of course remains the same. The total mass-travel, assuming maximum tire deflection and strut deflection occur together, for an 18 in. free drop in this case, is 18 in. + 4 in. + 6 in. = 28 in. The total energy to be absorbed is 28 in. × 4000 lb. = 112,000 in-lb. If the tire absorbs 40,000 lb., the strut must absorb 72,000 in-lb. Since the travel of the center of gravity due to the stroke of the oleo is 4 in., the average resistance of the oleo for the full stroke of 4 in. must be 18,000 lb. If the average resistance is again assumed as 80 per cent of the maximum, the maximum will be 22,500 lb. Based on the load of 4,000 lb., this is a factor of 5.62, which exceeds the 5.33 allowed.

The analysis of these two conditions demonstrates what happens when the shock-absorber leg is located so that the effective stroke in the direction of the center of mass travel is greatly changed by the attitude of the airplane in landing. If these two conditions represent the extreme variation of the airplane landing attitudes, then an obvious solution is to change the geometrical arrangement of the gear until the effective strokes under the two conditions are more nearly equal. While the one condition shows too high a factor, the other shows a factor considerably below that permitted. A rearrangement of the gear should bring the effective stroke for both types of landings such that the developed load factors for each condition probably will be within the maximum permitted, provided that the shock strut develops the same or nearly the same efficiency undér the two conditions. If it is not possible to make this arrangement, then the stroke must be increased until the both requirements can be met.

This analysis emphasizes two things, one of which is under the control of the designer and the other under the control of those responsible for fixing requirements. The first is that the location of the shock strut is very important in its effect on the possibility that the shock-absorber system will meet the

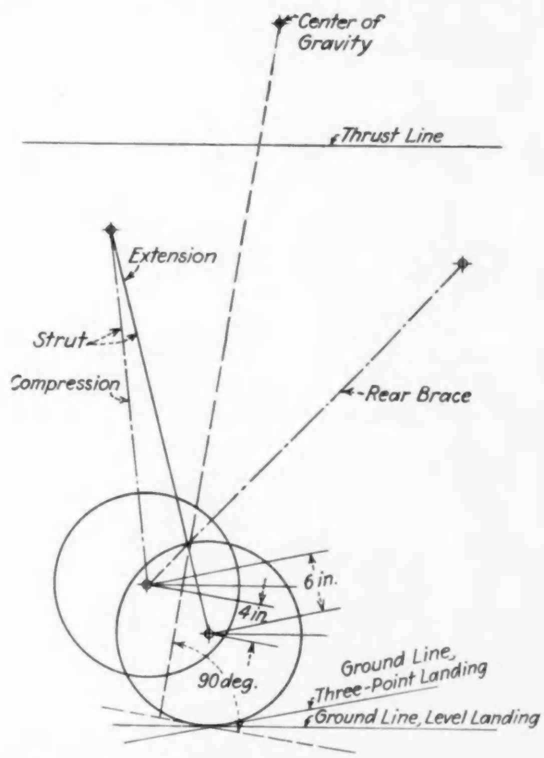


Fig. 1—Diagram of One Type of Landing Gear, the Axis of Rotation Being at Right Angles with the Longitudinal Axis of the Airplane

requirements of both the three-point and the level-landing conditions. The other point is that, in this arrangement of gear, the addition of the drag component in the level-landing conditions so modifies the direction of the reaction that the effective stroke is reduced to two-thirds of the original stroke. Part of this is of course chargeable to the geometrical arrangement of the gear. While the strut probably can be relocated or increased in stroke to take care of these two conditions, it is quite possible that actual operating conditions may introduce reactions and unit loads outside of this range that should be considered. A thorough investigation of the possible landing conditions that may be met should be carried out and a decision made as to the ones which the design must meet. Inasmuch as the gear in all probability cannot be designed to give equally good performance in all types of landing, the operating personnel should be informed regarding the limitations of the gear so that it may know how to get the best performance out of it.

It is now proposed to study the analysis just made from the standpoint of tire size and tire rebound. From the mathematical standpoint, and under the assumption made that the maximum tire deflection and strut stroke will occur simultaneously, all that increasing the tire size does is to increase the amount of energy that may be stored in the tire, and decrease the amount that must be absorbed by the strut. This permits several possibilities. A less efficient strut may be used, a shorter-stroke strut of the same efficiency may be used, or a lower load-factor may be developed provided it is not lower than that required to get the complete deflection out of the

Fig. 2—Set-Up of a Drop-Test Rigging for Testing a Strut and a Tire Alone

tire, and also that the strut is redesigned to take advantage of the increased capacity of the tire. In any event, there does not seem to be any reason to reduce the drop-height requirement if a larger tire is used, as was at one time proposed. The tire is a part of the shock-absorber system, and any benefit derived from it should show up in a reduction in requirements for the other members of the shock-absorber system, but the requirement for the entire system should remain unchanged.

The assumption that the maximum tire deflection and strut stroke occur simultaneously does not give the best performance, although it does theoretically result in the lowest load-factor with a given tire and strut stroke. The objection is that such a design will result in considerable bounce from the tire. If the tire is completely deflected at the end of the strut stroke, it has stored in it a lot of energy which it cannot dissipate. A pneumatic tire has a capacity of dissipating only about 10 per cent of the energy stored in it. It must get rid of this energy by lifting the mass of the airplane back into the air. In the case given, if the tire absorbs 40,000 in-lb., and dissipates only 4000 in-lb., it stores 36,000 in-lb., which is sufficient to lift the airplane 9 in. If the strut is completely compressed under this condition of full deflection of the tire, it acts as a solid member and the tire may actually lift the airplane the full 9 in. If, however, the strut still has an inch or so of stroke to travel, this stored energy in the tire may be given up to the strut as it moves downward, and it will be dissipated by the strut if it is provided with an energy-dissipating unit. To bring about this condition, the resistance developed by the strut must drop off near the end to an amount which will allow the tire to expand, thus giving up its energy. This is a highly desirable feature in that bouncing is reduced to a minimum.

The mathematical analysis is affected as follows: Instead of assuming that the tire will be deflected 6 in. at the end of the stroke, it will be assumed that it will be deflected 5 in. (The actual deflection will be determined in the design by the load developed at the end of the stroke by the taxiing spring provided in the strut.) At this deflection, the energy stored by the tire will be 26,000 in-lb. instead of 40,000 in-lb.

For the three-point analysis, the total mass travel will be 18 in. (drop) + 6 in. (strut-stroke movement) + 5 in. (tire deflection) = 29 in. The energy to be absorbed will be 29 in. × 3600 lb. = 104,400 in-lb. Crediting the tire with 26,000 in-lb., the strut must absorb 78,400 in-lb. instead of 68,000 in-lb. as before. Since the resistance of the strut must drop off near the end, the average resistance should not be as large a percentage of the maximum resistance as in the other strut. Assuming the same percentage, however, the load factor will be $78{,}400/(3600 \times 6 \times 0.80) = 4.4$ instead of 3.94. This is to be expected, since the resistance must be greater in the early part of the stroke to compensate for the drop off at the end of the stroke, and also the lesser amount the tire absorbs. The increase in maximum factor is more than compensated for by the reduced bouncing that will result if part of the energy initially absorbed by the tire is given up to the strut. Such a result cannot be achieved by a straight-displacement strut such as a spring or straight air-type, since the load must constantly increase to the end of the stroke in these types. Some type of energy-dissipating device functioning on the compression stroke, such as an hydraulic unit, must be provided.

The simple mathematical analysis just given may be applied to any proposed system. If the results of the analysis show that it is possible to meet the requirements with the

arrangement, the next step is to make sure that the shock strut will absorb the energy required. There should be no question about the capacity of the tire. The energy-absorption capacity of a shock strut may and should also be computed. This will show its limitations, but only an actual test will show whether or not the unit does what analysis shows it is possible to do. Obviously, it is futile to test a unit which can be shown by analysis to be inadequate. Further, the analysis shows just what may be expected, thereby eliminating gross errors which sometimes creep accidently into tests.

These check tests are usually of the drop-test type carried out with the entire gear or with the strut alone. Where the strut is mounted in the structure in such a way that it carries compression loads only, which are approximately the same load as is on the wheel or one-half the load if dual struts are used, a test of the strut alone represents working conditions quite accurately. If the geometrical arrangement is such that the load on the strut varies considerably from the load on the wheel, or it is subject to high bending-loads, the most satisfactory test is made by dropping the gear itself or an adaptation giving correct geometrical arrangement.

The most vital and difficult part of the drop test is the accurate measurement of the developed loads. Many times in the past drop tests have been made where the only criterion was whether any part of the gear failed or not. This only means that the particular part tested was strong enough to stand the loads developed, but does not indicate what might happen to adjacent parts. Sometimes, to the fact that no failure occurred is added the statement that no rebound was noticed. This means that the strut absorbed and dissipated sufficient energy early in the stroke, but does not mean that the resistance did not exceed the maximum permitted.

In recent years, instruments for measuring loads have come into use. Several forms of a time-space recorder from which decelerations can be obtained graphically, have been developed. Fig. 2 shows a test rig in use at the Bendix Products Corp. plant, with which a dePort-type time-space recorder is used. The arrangement when the strut is tested separately is shown, but with its tire or an equivalent one. Fig. 3 shows the same rig with a set-up representing a drop of the entire gear. It is with test equipment such as this that the actual performance of the complete system or its parts may be checked.

So far, only the effect of impact landings on the gear has been studied. Before discussing the system from the taxiing standpoint, it will be well to see if any recommendations regarding the gear can be made from these considerations. In view of the serious detrimental affect of a much-shortened effective-stroke resulting from one type of landing over another, it is evident that the shock strut should be located in the gear so that this change is reduced to a minimum or that sufficient stroke be provided for the worst condition. A second important consideration is that enough stroke should be provided to permit the energy-dissipating unit, which is usually the shock strut, to dissipate a part of the energy initially stored in the tire to reduce the amount of rebound from the tire.

The performance of a shock-absorber system while the airplane is taxiing across the field was formerly considered to be of minor importance. The speed was simply reduced if ground conditions were bad. With the speeded-up schedules required of air transports in recent months, much more attention has been given to this condition. At present, no definite requirement regarding performance is set up by our Government agencies. Transport operators specify comfort for their passengers and no hammering action on the structure. As in the impact condition, the tire and strut are the two units that must do this work. Further, it is the same tire and strut that are used to absorb the large amounts of energy developed in a hard emergency-landing that must also provide the comfort required for high and low-speed travel over all kinds of landing fields. This is not an easy specification to meet. Unfortunately for the designer, the more important of the two requirements, namely, that of safely meeting the hard impact of an emergency landing, is not often brought to the attention of the operator, while the lesser one is met on every landing and take off. The merits of the gear are thereupon quite likely to be judged by the performance under the taxiing conditions.

When considering the structural arrangement of the gear and its relation to taxiing, one is reminded that airplane under-carriages have had the much-advertised individual-springing or "knee-action" characteristics for several years. The advantages claimed for this type of chassis on the automobile may also be claimed for it on the airplane.

Each structural unit carrying the wheel and strut usually rotates about some hinge axis during the oleo stroke. The axis used affects taxiing characteristics. If the hinge line is parallel to the fore-and-aft axis of the airplane, careful attention must be given to the toe-in conditions of the wheel, as it is quite possible that large added loads may be introduced by the tendency of the wheel to pull outward or inward during this rotation as the airplane taxies on the field. This has been known to result in a stiff-riding gear, because the strut has been completely compressed or extended by this action.

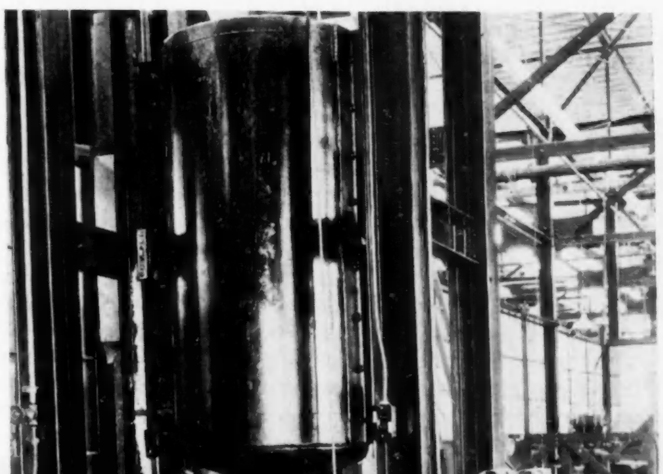


Fig. 3—Set-Up of a Drop-Test Rigging for Testing Both Struts and Tires

Another factor affecting taxiing will be brought out if the analysis made in the previous section for the three-point landing and the level landing with drag load is carried far enough to determine the loads in the shock strut for these two conditions. This will show that, for this arrangement, while the load on the strut in the three-point landing is 3600 lb., it is only 2500 lb. for the level landing with drag-load condition. Here the drag strut carries a much larger load. If the shock strut is provided with a taxiing unit, this change of load will result in giving considerably different characteristics for the two different conditions. Therefore, any adjustment in location of strut which can give approximately the same load on the strut in both attitudes would improve this condition.

Besides the effects of the structural arrangement, the shock-absorber units themselves must be considered. Much has been claimed for the large-capacity tire from the taxiing standpoint. Considered purely from the shock-absorption angle, the total capacity of the tire and strut together and the distance through which they travel should be the criterion. A low-capacity tire, together with a high-capacity long-stroke strut, should be equivalent to a high-capacity tire with a shorter-stroke strut. The capacity and stroke of the strut must be its spring capacity and stroke and not its capacity under impact conditions. In some types these are different.

Some installations have been made in which the tire has provided the only taxiing spring. When an oleo was also used, it functioned only on the impact of landing. Tires usually are completely deflected, if inflated to the correct pressure for the load carried, at a load factor between 3 and 4. Tests made by the National Advisory Committee for Aeronautics indicate that the loads developed in taxiing with conventional gears rarely reach 3 factors. From the load standpoint, then, a tire alone should be sufficient, since a nearly 4-factor load is necessary to make it bottom. One difficulty that arises with the tire alone is that it is an undamped spring. Since its capacity for dissipation of energy is low, it tends to oscillate for a long period unless a damping action is introduced, and this must be outside of the tire itself. Sometimes the oscillation period of the structure is such that the natural vibration periods oppose each other and damp out the oscillations. They might, on the other hand, synchronize in such a way that the oscillation would be increased. Further, a prolonged period of oscillation increases the change of the impact coming from the unevenness of the ground synchronizing with the oscillation in such a way that a high load may be built up.

When, however, the springing action of the tire is supplemented by a taxiing spring in the strut, many advantages result. The increased capacity and stroke permit reducing the size of the tire if the large size is not needed for other purposes. While this is an important advantage, the possibility of using the strut to damp out the natural oscillations of the tire is equally important. The structure of a well-designed strut provides means of snubbing the strut so that its natural period of oscillation may be changed to oppose that of the tire. The number of oscillations need not be more than three or four, and the last ones will be of small magnitude. The advantages of this are evident. A third advantage in having a taxiing spring in a strut is its ability to "keep the slack out of the system." If a spring is not used, a slight bounce may cause a small extension of the strut. Usually, the strut without the taxiing spring does not have any capacity of absorbing energy under a low-stroke velocity, and the strut bottoms noisily under the next compression. A taxiing spring prevents this.

Laboratory tests for checking the effectiveness of the shock-absorbing system under taxiing conditions have not been standardized. The drop-test rig having a reversible and continuous space-time recorder, such as is shown in Fig. 2, will show the oscillation period and indicate how effectively this is damped out. Study may show how some of the elaborate tests applied to determine riding comfort in automobiles may also be used in connection with the airplane landing-gear.

Our analysis of taxiing requirements has shown that the landing-gear designer should, if possible, make his gear so that no bad toe-in conditions of the wheel will make the gear stiff and that the normal loads on the shock strut will remain as nearly constant as practicable under the various possible airplane attitudes in taxiing. In addition, the shock strut should be designed to damp out the natural oscillations of the tire. Nothing definite has been set up regarding the actual energy-absorption capacity desirable for taxiing purposes for a given type of airplane. As a suggestion, a table of considerable interest could be prepared, giving, for several airplanes, the weight of the airplane, the total energy-absorption capacity of the tire and taxiing-spring unit of the strut under, say, load factors of 1, 2 and 3, and the total deflections of these two units under these same loads. This table would indicate existing practice. At present, our design assumption is that the greater the springing capacity and stroke are, the better the taxiing will be. This is limited, however, by the possibilities of trouble from big deflections on narrow treads in causing ground loops and other related difficulties.

The tail-wheel shock-absorbing system has much the same function to perform as that of the main gear and, therefore, most of the statements just made may also be applied to it. From the impact-loading standpoint, as the tail wheel only comes into action under the three-point-landing condition, the designer does not need to be concerned about changes in effective stroke, due to varying landing attitudes. He must, however, apply the same type of mathematical analysis to determine the minimum stroke at the wheel due to the oleo that is required, and then so locate the strut in the linkage that this stroke will result. Further, adequate stroke should be provided to permit absorbing most of the energy initially stored in the tire, for rebound from the tail is very undesirable. In general, there has been a tendency to provide insufficient oleo stroke for tail-wheel installation, and to let the fuselage structure absorb part of the energy.

A study of the unit, keeping the general requirements for taxiing in mind, leads one to question seriously the advisability of using the small-diameter wheels that have been common on recent designs. While the argument that the lack of shock-absorbing capacity in the wheel may be made up by using a longer oleo-stroke is correct so far as that point is concerned, the difficulty comes in applying the load in such a way that the shock absorber has a chance to function. Taxiing over rough fields, especially at high speed, introduces high loads applied parallel to the ground line. If the wheel diameter is small, the force may be applied as a pull on the axle which puts a high tension or bending load on the struts. Under this loading, in the type of gear in which the wheel is carried as a part of the lower end of the shock strut, the strut has to function purely as a structural member under high bending instead of functioning as a shock absorber. Where the wheel is carried on a rotatable truss hinged in such a way that the wheel can move backward and upward, thereby compressing the strut, which is mounted between this truss and the main structure, the strut can then function as a shock absorber if the wheel diameter is not so small

that it will not roll over the obstruction. There is little doubt but that the design should provide a tail wheel with a diameter adequate to roll over the type of obstructions that will be met, and that the installation should be such that the wheel can move both backward and upward under load.

The damping feature of the strut mentioned in connection with the landing gear to eliminate oscillation is probably more important for the tail strut, as the oscillation is more noticeable when applied so far from the center of gravity. In fact, a good shock-absorbing system for the tail gear is just as important under both impact and taxiing as for the main landing-gear.

Discussion

Research Needed On Orifice Design

—C. J. McCarthy
Chance Vought Corp.

THE general adoption of an hydraulic shock-absorbing member has made possible a vast improvement in landing-gear design. I recall that, when oleo shock-struts were just coming into common use in service-type aircraft, there was a lively debate whether or not military training-planes should continue to be fitted with the old rubber bungee shock-absorbers, on the theory that the new oleo-gears made it too easy for the student pilot to land his airplane.

Mr. Johnson does well to emphasize the importance of studying the geometry of the landing gear carefully to determine the best location for the shock-absorbing unit. The problem, as set up by the prescribing agencies, requires the designer to work out a landing gear which will have adequate shock-absorbing qualities to withstand a specified drop-test with the airplane in any attitude from a three-point landing to the two-point position, with the resultant force on the wheels acting through the center of gravity of the airplane. He points out that, in a conventional V-type landing-gear, locating the oleo member in the front strut is a favorable arrangement for the three-point-landing condition but is not at all good for the two-point condition, as the effective stroke is reduced one-third; in other words, for a given height of drop, the maximum acceleration is increased nearly 50 per cent.

The two-point position is a convenient one to use in stress analyses because the forces are in equilibrium, but I question if this is truly representative of service conditions, because many successful types of landing gears have been built with the oleo unit in the front strut and with a strut arrangement which gives relatively short effective oleo stroke for the two-point position. I believe it is quite probable that entirely safe landing-gears will result if the range of attitudes required for drop test were reduced to cover from the three-point position to a point about half way between it and the two-point attitude.

Under present requirements, to keep the length of travel of the oleo unit within reason, there is a strong tendency toward using a fixed V-structure with the oleo unit mounted in the V. With this arrangement the effective stroke of the oleo unit is substantially the same for any attitude between the three-point and two-point positions. It does, however, introduce bending into the oleo cylinder, which requires the use of liberal bearing surfaces; but it involves no unsurmountable difficulties, as the use of many single-strut landing-gears amply shows.

One would wish that Mr. Johnson might have devoted some space to discussing the design of the orifice of the shock-absorbing unit. To obtain a high-efficiency unit, one must ordinarily use an orifice which varies throughout the stroke. The size of the orifice is most conveniently controlled by the use of a tapered metering-pin, and the load-deflection curve or indicator card for the unit is very sensitive to minute changes in the dimensions of the metering pin.

Very few data have been published on this subject and, while a rough approximation of the shape of the metering pin can be made by calculation and comparison with the pins of gears previously tested, the only final recourse is to make a drop test of the landing gear, working out the shape of the metering pin by cut-and-try methods, which sometimes is a lengthy and laborious process.

Preliminary tests of the oleo unit alone are helpful, but the final design should be checked by tests of the complete landing gear, including the wheels and tires, as there is usually enough difference to require modifications in the shape of the metering pin. This problem is one which well deserves more thorough research than the airplane designer can afford to give to it.

Tire-Characteristic Control Of Tire-Unit Frequency

—William C. Peck
National Advisory Committee for Aeronautics

THE efforts of Mr. Johnson toward creating a better understanding of the requirements for an aircraft landing-chassis, the clarity with which these requirements are emphasized and the general comprehensive treatment given the subject, are to be commended.

The mathematical analysis of the absorbed-energy distribution in a landing chassis is more complicated than the simple analysis would indicate. The simple analysis will, however, amply serve to point out the possibility of a tire-strut combination absorbing a required amount of energy prior to dynamic tests on the combination.

Mr. Johnson, by his assumption that the maximum tire-deflection and strut stroke occur simultaneously, has treated the landing-gear shock-absorbing system as a unit of one frequency having two energy-absorbing reservoirs. In actuality, the conventional landing chassis acts as two separate but interdependent units—tire and strut—with different natural frequencies. The relative frequencies of these units influence the distribution of energy in the shock-absorbing system and are of prime importance in the performance of the landing chassis during ground runs.

The control of the frequency of the tire unit by the tire characteristics—the magnitude and geometric arrangement of the unsprung mass of the landing chassis, that is, the portion of the chassis between the ground and the upper section of the strut—has not been touched upon. Mr. Johnson, on the basis of his assumption of one frequency for the system, is correct in saying that "from a mathematical standpoint . . . all that increasing the tire size does is to increase the amount of energy that may be stored in the tire." Under actual work-

ing conditions, however, an increase in the tire size—with the usual proportionate decrease in inflation pressure—will lower the frequency of the tire unit. With the lowered frequency, short and sharp impacts, as encountered in ground runs, will be more completely absorbed by the tire without movement of the strut—in other words, without transmission of the impact to the aircraft structure—than with the smaller and higher-pressure tire. The general performance of the landing chassis in ground runs will be consequently improved with the use of the larger tire.

The statement made in the first paragraph that "in the event of a landing hard enough to cause some structural failure . . . the failure should be in the landing gear," is questionable. This statement would be generally accepted if it could be conceived that the failure would be confined to the landing gear. In most cases where landing-chassis failures occur, however, they are followed by extensive and severe damage to other structural parts of the aircraft. It would therefore appear to be more desirable to provide a landing chassis of such strength that, in a landing of sufficient severity to cause a structural failure, the primary failure would occur elsewhere than in the chassis.

Importance of Progressive Drop-Test Emphasized

—R. D. MacCart
Bureau of Aeronautics

MR. JOHNSON has so completely covered his subject and so ably presented the facts, that I find difficulty in adding anything of value and find nothing to criticize. At the risk of repeating what has already been said, I would like to emphasize the importance of the progressive drop-test up to the specified height and at various angles of thrust line to the horizontal. The progressive drops serve to represent the action of the landing gear during the later as well as the initial impacts such as occur during fast landings on rough fields, and the several angles of drop serve to represent the various landing-attitudes. For satisfactory operation during the progressive drops, as well as for proper coordination of action of tire and shock struts, a metering orifice is considered to be advisable if not essential and, for satisfactory operation under the various angles of drop, a suitable shock-strut location is necessary.

In this connection it should be realized that the specified height of drop is intended to represent the extreme rather than the usual vertical velocity. It should also be emphasized that, although the shock-strut wheel-combination drop-test results are of value in the case of all landing gears, and of great importance in some, the real criterion is the performance of the gear as a whole when installed on the fuselage or on a jig representing the fuselage. Because of the different behavior of the shock-absorbing system when incorporated in the landing gear, except in special cases it is unwise to attach too much importance to the former test.

The landing-gear shock-absorbing systems, both undercarriage and tail wheel, have shown very great improvement during the last two years in airplanes constructed for the Navy. This has been brought about by the high specified drop-height, the requirement that the structure be designed for the acceleration actually developed, and the drop-height measurement being measured at the center of gravity of the airplane. The first two requirements bring about an efficient shock-absorbing system—low accelerations—because of the reluctance of structural designers to increase the structural weight to take care of the high loads imposed on the structure by the inefficient shock-absorbing systems, and the effect of the third requirement is to provide a reward for the use of a long-stroke shock-system in reducing the distance between the ground and the bottom of the wheels. Since it is the distance through which the center of gravity drops that determines the energy to be absorbed, it seems logical for this as well as for other reasons to specify the height of drop with reference to the center of gravity of the airplane rather than with reference to the bottom of the wheel.

I would like to point out that any system of measuring the accelerations at the tail wheel when dropped on the fuselage by rotation about the undercarriage axle will give data that cannot be tied up with applied loads used in design, as the drop in this case is mostly rotation about the axis of the main landing-gear wheels and the impact loads are applied at considerable distance from the center of gravity of the airplane. It is customary to design the tail wheel for a given load-factor applied to the static loads as measured when the airplane is in the three-point attitude with the designed gross weight of the airplane applied at the center of gravity of the airplane.

To tie up this method with the actual accelerations as determined from the drop test, it is necessary that account be taken of the moment of inertia of the airplane, the distance from the center of gravity to the main landing-gear axles, the distance from the main landing-gear axles to the tail wheel, and the location of the recording or measuring device from which the accelerations are determined. This may appear rather complicated; but, after preliminary calculations are made and the distances measured, for any height of drop the load factor will be $aK + 1$, where a is the measured and calculated acceleration at or near the tail wheel and K is a constant for that particular airplane. The load factors thus determined bear a direct relation to the load factor applied to the static load in the three-point attitude.

Practical Improvements In Design Suggested

—R. W. Ayer
Stinson Aircraft Corp.

UNTIL recently, we have always felt that the Department of Commerce requirements for an 18-in. free-drop were meant practically exclusively to take care of a case of hard landing caused by misjudging the distance to the ground, either in a plus or minus direction. The set-up seemed quite logical to us, until a series of miscellaneous observations were made on various types of shock-absorber struts during a normal hard-landing. It was found that, when using a strut of normal design—designed to bottom at, roughly, five load-factors—it was almost impossible to get a sane pilot to make a bad enough landing to deflect the shock-absorber strut more than half-way.

These observations were made not only on the usual type of spreading-type landing-gear, but also on the recent type in which the wheel imparts its load directly into the shock absorber. This led us to the tentative conclusion that, except in the case of a crash, the loads on the shock absorber rarely approach the condition for which they are designed. We excuse this discrepancy on account of the fact that, even in a

very bad landing, the wings are still contributing a very considerable lift and, therefore, the acceleration in the drop is nowhere near approaching that of a free drop.

At the same time, however, it was observed that, when the airplane was slowed down sufficiently to remove a large part of the lift from the wings, the loads on the shock absorber ran extremely high, and sometimes the shock absorber on a rough field would actually bottom at speeds as low as 10 m.p.h. We then decided that shock absorbers were lacking more in taxiing qualities than they were in their ability to absorb the shocks from landing.

It then appeared necessary, to get a strut that would taxi nicely and still meet the Department of Commerce requirements, to have a great excess of travel and a relatively inefficient strut for the travel given, as far as energy absorption is concerned. This was done and, although no quantitative results are available, the strut is giving us the best taxiing qualities that we have as yet experienced and will still absorb the energy of the hardest landing very satisfactorily without bouncing.

We have found that giving the tail-wheel shock-absorber struts a chance to move backward at the early part of the stroke is one of the most important factors giving a soft-taxiing tail-wheel. In fact, we are of the opinion that the backward motion of the wheel should roughly equal the upward motion of the wheel throughout the total stroke. Since this can be done easily without running into difficulties with shimmy, we believe it will become standard practice.

In general, the tail-wheel oleo should be provided with the same length of stroke as the main-wheel shock-absorber; but we believe that, if the backward motion is furnished, some cutting down of the tail travel can be permitted and still obtain easy riding.

It has been our experience that, in general, the increase of unsprung weight in an airplane landing-gear improves the taxiing quality, particularly where a high-capacity tire is provided. Although this seems rather contradictory to automobile experience, it may be due to the fact that the unsprung weight still has a very small ratio to the sprung weight as compared with that of an automobile.

We believe also that a little excess of unsprung weight not only causes the tire to absorb more of the small shocks but also tends to extend the oleo a little quicker and make the wheel follow the ground more closely. The packing on most oleo struts as they are now furnished has to be so tight to prevent leakage that it very often happens that the weight of the wheel alone will not extend the struts much beyond half-way, unless the strut happens to be a very high-pressure type with the normal taxiing position well extended.

In conclusion, we believe that the industry has been gradually changing its point of view with regard to the importance of a good shock-absorbing system, and that this is due in a very large measure to the researches into the subject that have been carried on by the two leading manufacturers of shock-absorbers, one of which Mr. Johnson represents. The mental hazard that is present in a pilot's mind when he knows that an airplane is a bit touchy on the ground is often sufficient to cause the pilot to be prejudiced against the airplane's other qualities. Conversely, an airplane which taxies soft on any kind of field and is entirely dependable with regard to its ground characteristics will often make the pilot feel very friendly toward an otherwise mediocre airplane. There seems to be no real good excuse, therefore, in compromising on taxiing characteristics of an airplane, since the structural difficulties in obtaining good characteristics are seldom large.

Close Cooperation Between Designers Is Advocated

—Theodore dePort
Wright Field, Dayton, Ohio

THE engineering problems of the shock-absorber designer are well illustrated in detail in Mr. Johnson's paper, which is recommended to be studied by airplane manufacturers and, especially, by the landing-gear designers.

It must be understood by those responsible for the landing-gear design that the best shock-absorber cannot function to its best advantage on a poorly designed landing gear. The mechanics of this part of the airplane must be such as to permit the shock absorber to do its work. It is a common occurrence that the airplane designer requires from the shock-absorber designer a strut which would have to be more than 100 per cent efficient to satisfy the load requirement. This, of course, is physically impossible.

It is not often realized that, given the mechanical conditions of the landing gear, including the stroke of the strut and height of drop, there is a certain minimum dynamic load that can be obtained no matter how efficient is the shock-absorber design. It happens quite often that this minimum dynamic load possible is above that allowable by specifications. A close cooperation between the designers of landing gear and the shock absorber at the beginning of the design should be maintained to obtain the best results.

Notwithstanding the popular opinion that large-diameter tires increase the capacity of the energy absorption, I agree with Mr. Johnson that this is not the case, especially when low pressures are used. If a change from high to low-pressure tires is made, the orifices in the strut piston should be altered also if they were correct in the first place and if the maximum efficiency is required. However, in many cases this will not be necessary, since the range of conditions is large under which a properly designed landing-gear shock-absorber unit will operate; but in any case, the stroke of the strut should not be shortened on account of larger tires.

Shock-Absorber Design-Problems

The ever-increasing speed of airplanes is directly responsible for the necessity for corresponding increase in the efficiency of shock absorbers. Taking for example an airplane of several years ago, say a Curtiss JN-4 with a high speed of about 80 m.p.h. and a landing speed of 40 m.p.h., its shock absorber consisted of a few rubber chords allowing a relative motion of the wheel axles to the fuselage of about 4 in. The efficiency of dynamic shock absorption of rubber or any other elastic medium is about 50 per cent of the theoretical maximum possible. This low efficiency, with the short relative motion of 4 in., was found sufficient to take care of landings at 40 m.p.h. with corresponding low vertical-velocities.

The vertical velocity of an airplane in landing is in approximately direct proportion to its landing speed, and the energy to be dissipated by the shock absorber is in proportion to the square of the vertical velocity. The result is that, if the high speed of an airplane is doubled, and assuming that its speed ratio is not improved, the landing speed will also be doubled. Then, the kinetic energy that must be absorbed by the landing-gear shock-strut is four times as great; therefore, if the shock-absorbing efficiency is not improved, the stroke or the relative motion of the shock absorber must be quadrupled, which, in the above example, will require a 16-in. travel

of the axles relative to the fuselage. This long travel is, in the majority of cases, impractical both from structural and aerodynamic reasons.

The advent of reducing the landing speeds by use of high-lift devices such as slots and flaps does not improve the matter but makes it worse. These devices, while decreasing the forward speed, also increase the angle of the flight path relative to the ground. The vertical velocity is a function of both of these factors, that is, the landing speed and the angle of flight path, and it happens that the reduction of landing speed does not compensate for the increase of the path angle, the result being that the vertical velocity at landing is even greater with the use of the flaps than without them.

To satisfy the landing condition of present-day airplanes, the requirement of the strength of landing gears and fuselage was increased to a permissible limit which does not entail a too-great increase in structural weight. The necessary improvement in shock-absorbing capacity without increasing the stroke was made by using correctly designed hydraulic struts.

The general laws governing the functioning of an hydraulic or oleo shock-absorber were well known for a long time, but only for relatively low velocity. When it is considered that the oil passes through the piston orifice of an oleo strut at a velocity as high as 300 m.p.h., it is easily understood that the accepted laws may not hold. For example, the tests conducted by the Army Air Corps showed that, under certain conditions, the hydraulic pressure is proportional to the square root of the velocity, while the accepted formula gives the proportion to the square of the velocity.

The Materiel Division at Wright Field conducted extensive tests on the flow of liquids through orifices to obtain basic data useful for the designer and developed a testing apparatus especially for this work; that is, for measuring dynamic forces in shock absorbers in the drop-testing jig simulating conditions present in the landing of an airplane. One of the first lessons learned was the effect of the tire on the functioning of the shock absorbers. This effect is so great that, if neglected in the design, the strut would be worthless. This introduction of the tire effect is exceedingly complicated because not only the functioning of the strut depends on the characteristics of the tire but also the functioning of the tire is altered by the action of the shock absorber, both of them forming a unit of definite characteristics. However, at present, the interdependence of the tire and shock absorber is known and the most advanced designs incorporate the feature of a specially designed metering pin which permits very high dynamic efficiency, as high as 93 per cent for the strut and somewhat lower for the tire and strut unit.

Another problem in shock-absorber design is that of the taxiing. The requirement for taxiing is for obvious reasons entirely different from that of landing; however, the same unit, the tire and strut and landing gear, serve for this purpose. Tires alone, and especially the large-diameter low-pressure tires, are good taxiing mediums; but they still must be supplemented by the actions of the strut. The inadequacy of the tire is supplemented by metallic springs and, in recent years, by compressed air in the strut. The springing action, whether furnished by a steel spring or by the compressed air, is another factor entering into the determination of the oleo design; especially as the compressed air produces a double effect on the kinetics of the shock absorber, one effect being due to the internal pressure acting directly on the piston, the other acting on the air-tight packings and producing considerable friction which further alters the condition of the hydraulic parts of the oleo strut.

The Materiel Division undertook the basic and original research work and determined the effect of different variables upon the hydraulic resistance of fluids. The following variables were investigated:

(1) Different shapes of plain circular orifices

(2) Varying dimensions of ring orifices which are formed when circular orifices are used with a concentric metering pin of constant or variable diameter

(3) Piston pressures

(4) Piston velocities

(5) Viscosity of fluid

(6) Piston leakage

(7) Temperature

(8) Change in fluid density due to emulsification of the fluid by the air during the operation of the strut

The results of these tests showed that these factors play important parts in the functioning of the oleo shock-absorbers and that the general hydraulic laws of flow of fluids through orifices must be modified accordingly.

The most important single factor contributing to the increase of efficiency was the discovery of mutual effect of the tire reaction in the functioning of the oleo shock-absorber, which resulted in the design of the "knobbed" metering-pin. The principal function of this type of pin is to retard or stop the motion of the piston in the oleo strut until the tire is deflected, and then to maintain the constant load in the strut throughout the largest possible part of the stroke consistent with the effect of the tire returning to the static position.

Here it may be recalled that the criterion for shock-absorber efficiency is the shape of the load-deflection curve. The maximum possible efficiency will be obtained when the graph is rectangular; that is, when the area under the curve for given maximum load and stroke is the maximum. This area represents the energy absorbed or dissipated, as the case may be. The desired condition is that the energy corresponding to the vertical component of the kinetic energy of a landing airplane is absorbed by the strut-tire combination with minimum stroke and without exceeding the allowable load-factors and without rebound.

The effect of increased shock-absorber efficiency may be noticed in the landing of modern airplanes. While in the past the pilots were very anxious to "feather out" their landings by careful leveling off near the ground, now they become more negligent and often fly the airplane "into the ground" and get away with it, thanks to good shock-absorbers.

Commends Paper and Suggests Additions

—G. A. Luburg
Curtiss Aeroplane & Motor Co., Inc.

MR. JOHNSON'S paper certainly is well written and unusually free from statements that would tend to arouse controversy. While this latter characteristic makes comment difficult, I would suggest that he might have gone a little bit further into the discussion on the effect of steeper gliding angles which are coming with the increased use of slots and flaps.

Also he might have recommended a taxi load requirement.

We believe that a change should be made in the present specification which penalizes long travel gears by measuring the height of the drop from the extended position rather than the taxiing one.

Discussion of Graves-Mougey-Upham Paper Brings Out Further Data on Winter Oils

THE S.A.E. 10-W and 20-W oils, which are classified in accord with their viscosity at 0 deg. fahr., were adopted for publication and trial in June, 1933. The paper, entitled Winter Oils for Automobile Engines, was published in the July, 1934, S.A.E. JOURNAL, pages 238 to 247 inclusive, and stated the results of the use of these oils during the winter of 1933-34, together with their advantages. The data presented showed that these oils are necessary for cold starting, provide adequate lubrication and improved performance, and give reasonable oil mileage.

In the discussion of the paper, published herewith, further data on viscosity-temperature characteristics are presented, the importance of oil-consumption efficiency is enlarged upon, and the adoption by the Society of the tentative 10-W and 20-W classifications is said to constitute a more controversial matter than that presented by the relative service-characteristics of high and of low-viscosity oils.

Oil Consumption Discussed Further

—D. P. Barnard
Standard Oil Co. (Indiana)

THE authors of this paper have expressed with clarity the views which are shared by a majority of those most thoroughly acquainted with the problems of automobile lubrication. While the paper deals with cold-weather-lubrication problems only, the matter is one of first-magnitude importance because of the fact that the difficulties attendant to cold-weather operation are those which strike directly at the basic utility of the automobile. As the primary purpose of the automobile is that of furnishing transportation in spite of the obstacles imposed by nature, those problems which limit its use become very considerably more important than those which deal only with variations in operating expense and the like. An inspection of the viscosity curves given in the paper shows almost at a glance the great desirability of assuring for winter use those oils which are described as 10-W and 20-W rather than those which are permitted under the older specifications.

In spite of the fact that assuring the possibility of starting and continued satisfactory operation in cold weather is certainly the most important consideration, it seems that the matter of oil consumption occupies, in the eyes of the public at least, almost, if not entirely, as important a position. Even if operating expense of the car is placed on as important a level as reliability of operation, it would seem that the feature of oil consumption has been greatly overstressed, as has been pointed out by the authors. It would seem that this point as brought up in the paper might be discussed a bit further.

If we consider that it is quite improbable that the indicated horsepower of an engine will be altered appreciably by oil-viscosity changes, it at once becomes obvious that the 3 per cent increase in fuel consumption which was observed could have resulted only from a corresponding decrease in brake-horsepower output. When we consider some of the expedients which are resorted to in order to obtain rather small increases in engine output, it would seem that such a decrease should not be accepted unless absolutely necessary. For example, in terms of present practice, a 3-per cent decrease in output would be equivalent to a drop of the compression ratio by an amount corresponding to three octane numbers on the fuel to be used. This is a fair proportion of the difference in price between gasolines of the different knock-rating grades. Actually, however, when the decrease in output results in an increase in internal engine friction, the matter becomes somewhat more serious than this as it is effective during the entire operating time of the engine, whereas losses in octane number are only noticeable during full-throttle operation.

It is interesting to note that, in connection with the values quoted, some observations have been made by our own organization which indicate almost exact agreement in relation between oil viscosity and fuel consumption. Further, in connection with some tests recently carried out on a number of motorcoaches, the low reserve-power of these vehicles brought home quite forcefully the effect of impairment of engine output in that, under one set of conditions, in which operation on a grade of something under 4 per cent was involved, the attainable speeds were 14.7 m.p.h. when an S.A.E.-50 oil was used and 18.1 m.p.h. with the 30 grade. Further, these observations were made, not at winter temperatures, but at an air temperature above 60 deg. fahr. Another and somewhat indirect confirmation of the power loss which must accompany the use of oils of high viscosity was obtained during the Speedway tests carried out by the Standard Oil Co. (Indiana) three years ago. Some of the data obtained in this

work were presented[1] at the Annual Meeting of the Society in 1932. The work at that time indicated that engine losses were effected by oil viscosity in exactly the manner that would be predicted from the laws of viscous flow and that the effects of viscosity and rubbing speed are exactly in accordance with theoretical requirements.

It certainly appears that the avoidance of excessive viscosity, at low temperatures in particular, should justify any reasonable expenditure; and, when it appears so certain that, within fairly reasonable limits, the advantages of increased reliability of operation which can be realized by the use of low-viscosity oils actually cost little or nothing in the long run, it would seem that the possible criticism of the use of such oils on the basis of consumption should not be a particularly serious one.

One possible minor criticism of the proposed system for designating oils for low-temperature service arises in the use of the A.S.T.M. Viscosity-Temperature Chart. This is not due to any possible error in the chart itself as, even if such discrepancies do exist, their effect upon the comparisons of different oils would be quite negligible; and, just so long as the same chart is used by everyone, no possible trouble from this source could ensue. It is assumed, of course, that reasonable intelligence will have to be used in the viscosity determinations themselves, as, for example, it is well recognized that the Saybolt instrument is not sufficiently accurate for this work when efflux times are lower than 50 sec. Further, there are some possibilities that certain materials may exhibit plastic properties, and, therefore, will have viscosities at low temperatures higher than would be predicted by extrapolation of normal-temperature data by means of the chart. At present, however, it certainly looks as though such effects could not amount to more than 30 or 50 per cent at 0 deg. fahr. This is almost insignificant in comparison with the ten-fold and larger viscosity-differences which now can and do exist under the present S.A.E. classification-system. It appears, therefore, that the method for describing 10-W and 20-W winter-oils, as proposed by the authors and which has worked so successfully on a tentative basis during the last year, should be wholeheartedly supported.

Summarized Data on Viscosity-Temperature Characteristics

—P. K. Frolich
Standard Oil Development Co.

WE have all listened with great interest to Mr. Mougey's presentation. After having heard this excellent review of the subject, it is hard to understand how we can have been so slow in adopting low-viscosity oils for ease of starting at low temperatures. The authors are to be congratulated on their successful efforts in this direction.

The authors point out that crankcase oil must perform the following three functions; (*a*) furnish adequate lubrication, (*b*) permit easy starting, and (*c*) give reasonable oil mileage. Of the evidence presented to show that light oils give adequate lubrication, the tests conducted with the Dayton Power & Light Co.'s bus fleet are the most convincing. Most of the other tests are merely reported as giving an apparently satisfactory engine condition.

Recently, Dr. Everett, of Penn State University, presented a paper before the convention of the Pennsylvania Grade Crude Association discussing the results obtained with various lubricating oils in his recent tests on the University's battery of four Dodge engines. The interpretation of these data in the light of our own experience leads to the conclusions that:

(1) Blowby is related to oil consumption.

(2) The greatest consumption is obtained with the oil of the lowest viscosity at 350 deg. fahr., the viscosity at this temperature having been chosen from work in the Standard Oil Development Co.'s laboratories as a representative index of oil consumption.

(3) The greatest wear occurs with the oil of lowest viscosity at 350 deg. fahr.

(4) The most sludge is formed with the oil of lowest viscosity at 350 deg. fahr.

This raises the question as to how low one may safely go in viscosity and yet obtain satisfactory lubrication.

As for the other two of the requirements, tests in our laboratories have shown that ease of starting and oil consumption are tied together and are dependent upon the viscosity-temperature characteristics of the oil. For convenience, the data have been summarized in Fig. 1. In the lower left-hand corner we have plotted cranking torque in foot-pounds as a function of viscosity at 0 deg. fahr. Starting with 62.4 ft.-lb., which, according to the authors' data, is the torque available at 0 deg. fahr., we find that this corresponds to a viscosity of about 20,000 sec. at that temperature. If we go up vertically from this point to the upper left-hand plot, we may now convert from viscosity at 0 deg. fahr. to viscosity at 210 deg. fahr. by the aid of a series of viscosity-index curves. If the viscosity index of the oil in question happens to be 100, we find that the corresponding viscosity at 210 deg. fahr. is 51 sec. By proceeding horizontally to the upper right-hand diagram, we may in similar manner convert viscosity at 210 deg. fahr. to viscosity at 350 deg. fahr., which latter is taken as representing average temperature of the oil film on the cylinder wall. Finally, the lower right-hand corner expresses the relation between the viscosity thus found and rate of oil consumption in quarts per thousand miles, as determined on a rear-wheel dynamometer with a 1934 Ford V-8 for speeds of 50 and 60 m.p.h.

Fig. 1 is very useful in that one can start with the authors' definition of easy starting and determine from the viscosity-temperature characteristics of the oil what the relative consumption will be. Granting that the torque available from the battery determines the maximum viscosity which can be tolerated for satisfactory starting under winter conditions, it follows from these data that the viscosity index of the oil has a profound influence on its rate of consumption. Thus, of two oils of 50 viscosity index and 120 viscosity index, which have identical viscosity at 0 deg. fahr. and therefore give the same ease of starting at that temperature, the one with the lower viscosity index would be consumed at a rate of 2.75 quarts per thousand miles, whereas the consumption of the higher viscosity-index oil would only be 1.65 quarts per thousand miles, when operating at 50 m.p.h.

The point we want to emphasize is that high oil-consumption is not just so much money thrown away—since, as the authors have pointed out, this may even be balanced in some

[1] See S.A.E. JOURNAL, May, 1932, p. 192; A Possible Criterion for Bearing-Temperature Stresses, by D. P. Barnard.

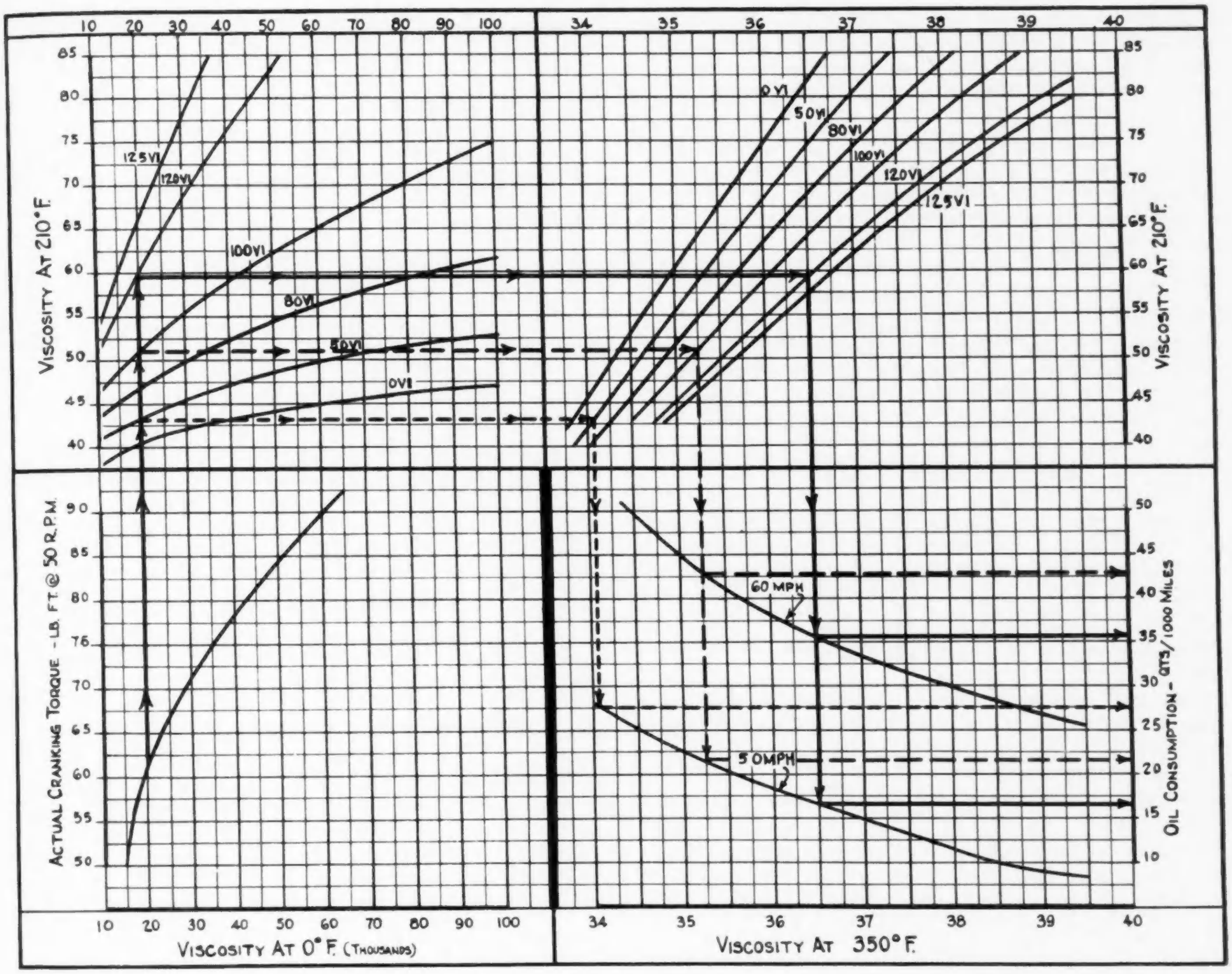


Fig. 1—Consumption and Cold-Starting Characteristics of Oils of Various Grades and Viscosity Index. Engine Data from 1934 Ford V-8 Tests

cases by a corresponding saving in gasoline—but rather that a high rate of consumption indicates; (*a*) risk of complete lubrication failure (particularly due to an unexpected rapid exhaustion of the oil supply on high-speed driving), (*b*) greater wear, and (*c*) more sludging.

Adoption of 10-W and 20-W Classifications Considered

—G. L. Neely
Standard Oil Co. (Calif.)

THE title of the paper, "Winter Motor Oils for Automobile Engines," may well have been named "The Relative Service Value of Low and High-Viscosity Oils." Viewing the paper from this aspect alone we may say that we are in strict agreement with the conclusions drawn in regard to the better lubrication obtained, the adequate engine protection afforded and the compensating effect of increased gasoline mileage as related to oil mileage when using low-viscosity oils. In addition, we would like to add that our experience has been that the use of light-bodied oils has resulted in much cleaner engines from the standpoint of deposits in the crankcase and in the combustion chamber. In view of the present interest in crankcase-oil stability, it is felt that the value of light oils in regard to general cleanliness of the engine should be considered an important item. Many references are presented by the authors showing that, without exception, light-bodied oils gave adequate engine protection under severe operating conditions. We may say that we, also, have obtained considerable data along this line which are distinctly favorable to low-viscosity oils.

It is believed that the relative value of light-bodied oils versus heavy oils has been agreed upon by the majority of the technical men of both the automotive and the petroleum industries and it is felt that, if a census were taken, it would be found that these technical men have for the most part used low-viscosity oils in their own cars for years. Nevertheless, the false economy of using heavy-bodied oils has never been brought forcefully to the attention of the consuming public. The absence of support from the automotive industry has in the past rendered such a policy futile, due to the feeling on the part of the general public that oil mileage is an index of oil quality.

The chief problem has been that differences in oil mileage between light and heavy oils are obvious to the car operator, whereas the compensating differences in gasoline mileages are not obvious. It has been concluded that, to re-educate the motorist in regard to viscosity oil-grade, he must first be told by the motor-car manufacturers that high oil mileage is not desirable if obtained by the use of highly viscous oils. The paper under discussion completes the record in so far as technicians of the motor-car industry have expressed themselves; but, for the motoring public to benefit from the advantages of light oils, the motor-car industry must uniformly recommend light oils. Positive action to this end would be an immediate blanket recommendation of S.A.E.-20 oil as the heaviest oil suitable for new cars, at least for the first 10,000 to 15,000 miles, regardless of the atmospheric temperature or type of driving. If this were done, we feel that the car owner would find that adequate oil mileage would be obtained with a light-bodied oil and that at the end of the 10,000 to 15,000-mile period there would be no need for using the heavier grades now recommended. However, a retroactive recommendation leading to the use of light-bodied oils in cars that have been driven 30,000 to 40,000 miles on heavy-bodied oils would not prove successful because of the comparatively poor oil mileage that would be obtained with the low-viscosity oils.

A consideration of the paper from the standpoint of advocating the adoption by the S.A.E. of the tentative 10-W and 20-W classifications constitutes a more controversial matter than that presented by the relative service characteristics of high and low-viscosity oils. While it is agreed that cranking at low temperatures is a matter of viscosity, there are several reasons why the adoption of the 10-W and 20-W oils should be rejected.

Attention is called to Fig. 2, in which the limiting cranking temperatures for oils of various grades and viscosity indices are shown. In this chart, the limiting cranking temperatures include the effect of temperature on battery output in that they are based on viscosities defined by straight lines passing through 40,000-sec. Saybolt at 0 deg. fahr. and 25,000-sec. Saybolt at —15 deg. fahr., using a rectangular plot. This is in accordance with data published by Mr. Mougey in the *Oil and Gas Journal,* March 19, 1931, page 30. It is of interest to note that this classification also provides —15 deg. fahr. as a cranking limit for the 10-W oils even though this limit is based on viscosity at 0 deg. rather than at —15 deg. fahr. A consideration of Fig. 2 clearly shows the degree of overlapping incurred by the use of the present S.A.E.-10 and S.A.E.-20 classifications and the tentative 10-W and 20-W classifications. It will be noted that all oils of 100 viscosity index or above may be designated as 10-W, 20-W, S.A.E. 30, 40 and 50 without reference to S.A.E.-10 and S.A.E.-20 grades, whereas a majority of the oils of viscosity indices less than 100 falling in the present S.A.E.-20 classification would require a continuance of the S.A.E.-20 classification for designation. This fact makes it possible for motor-car manufacturers and distributors of 100-viscosity-index oils to recommend 20-W oils for winter use and S.A.E. 30 or 40 for summer use. By such recommendation on the part of the manufacturer, that group of oils shown in area *A,* Fig. 2, of the S.A.E.-10 and S.A.E.-20 classification may become obsolete in so far as motor-car manufacturers' recommendations are concerned, in spite of the service value of these oils.

In connection with the foregoing remarks, attention is called to an article by Mr. Mougey published in *Automotive Industries,* April 28, 1934, page 532, entitled 10-W and

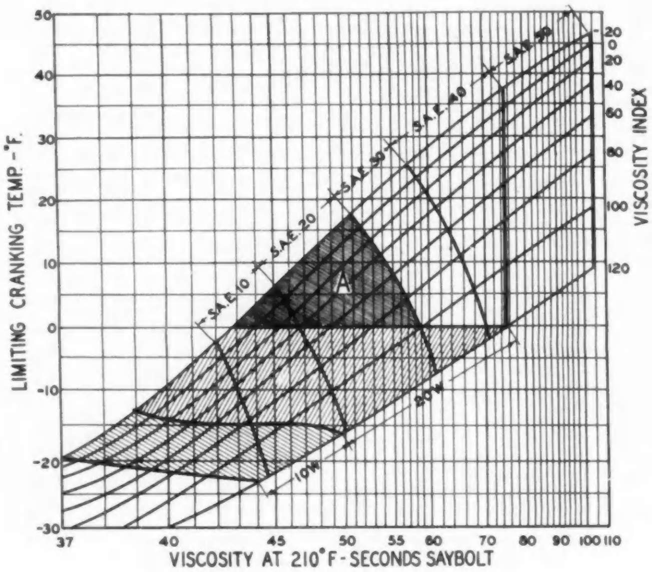


Fig. 2—Engine-Lubricating Oils—Proposed Changes in S.A.E. Classifications

It should be noted that limiting cranking temperatures include effect of temperature on battery output in that they are based on viscosities defined by a straight line passing through 40,000-sec. Saybolt at 0 deg. fahr. and 25,000-sec. Saybolt at —15 deg. fahr. (rectangular plot)

20-W Lubricants Expected to Obsolete S.A.E. 10 and 20 Oils. Mr. Mougey concludes as follows: "We are particularly anxious to have this whole matter cleared up before the June meeting of the S.A.E. for the reason that at this meeting some action will be taken for officially approving the use of the letters S.A.E. in connection with the Classification 10-W and 20-W. The probabilities are that these oils will be known after the summer meeting as S.A.E. 10-W and S.A.E. 20-W. It is my belief that the present S.A.E. 10 and S.A.E. 20 will become obsolete, but whether or not S.A.E. 20 will be repealed at the June meeting or even for several years is a question. It is necessary to make changes slowly where very large commercial interests are involved as in the case of proper oils for winter starting." We are in agreement with Mr. Mougey's conclusion that an overlapping classification such as would be effected by the official adoption of 10-W and 20-W grades can have but one ultimate result, and that is further modification to eliminate this overlapping in classifications.

The adoption of the tentative 10-W and 20-W grades would, in effect, result in the elimination of the present S.A.E.-10 grade as such, and would adversely affect the sale of a large group of the low-viscosity-index S.A.E.-20 oils. Certainly, any move which leads to a specification whereby a large group of oils might become obsolete as predicted by Mr. Mougey is not acceptable, and particularly so where the oils rendered obsolete are especially suitable for engine lubrication, and are widely accepted by the public.

In the tentative classification of the 20-W oils, viscosity of 40,000 sec. Saybolt at 0 deg. fahr. was used as the upper limit. It is of interest to note in Fig. 2, that the lines defining the upper limit of the 20-W oil, and the lower limit of the S.A.E.-30 grade, intersect at exactly 100 viscosity index. Therefore, the choice of a limiting viscosity of 40,000-sec. Saybolt permits the designation of only 100-viscosity-index oils or higher by the 10-W, 20-W and S.A.E.-30 grades without utilizing the S.A.E.-10 and S.A.E.-20 classifications. It is noted in Table 3 of the paper that a value of 26,000 was

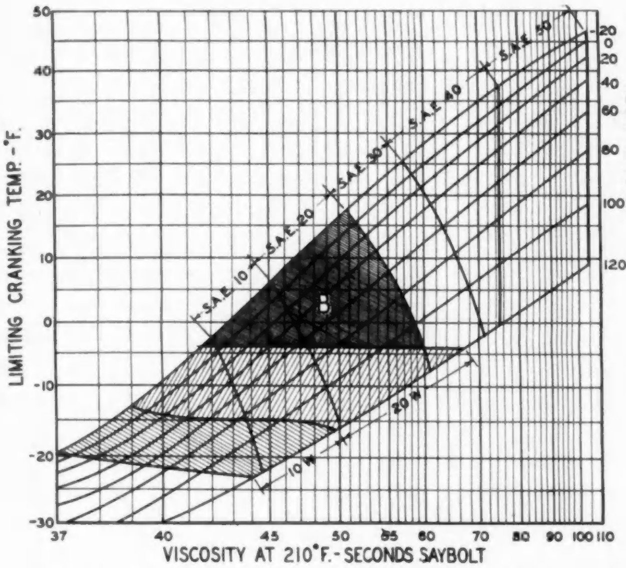


Fig. 3—Engine-Lubricating Oils—Proposed Changes in S.A.E. Classifications

It should be noted that limiting cranking temperatures include effect of temperature on battery output in that they are based on viscosities defined by a straight line passing through 40,000-sec. Saybolt at 0 deg. fahr. and 25,000-sec. Saybolt at —15 deg. fahr. (rectangular plot)

obtained by Blackwood and Rickles and a value of 30,000 by Larson. Many companies throughout the country commonly use 30,000 as a limiting cranking viscosity.

Let us see then what happens if the upper limit for the 20-W oil is based on 30,000 rather than on 40,000 sec. Saybolt. This is shown in Fig. 3. The reduction in viscosity limit from 40,000 to 30,000 constitutes a reduction in cranking temperature of approximately 4 deg. fahr., and extends that area of the S.A.E.-10 and S.A.E.-20-grade oils, which would be rendered obsolete, in the opinion of one of the authors, by the adoption of such a classification with such an upper limit. It is thus seen that 100-viscosity-index oils falling between 54 and 59 sec. at 210 deg. fahr. would likewise become obsolete, whereas oils of 0-viscosity index falling in the "obsolete" class would be reduced about 1-sec. Saybolt at 210 deg. fahr. A survey of the oils sold on the Pacific Coast in the present S.A.E.-20 classification shows the large majority of 100-viscosity-index oils of the present S.A.E.-20 grade fall between 54 and 59 sec. at 210 deg. fahr. In other words, practically all oils now available in the S.A.E.-20 classification would likewise be rendered obsolete. We feel sure that no petroleum manufacturer would be willing to render obsolete oils falling in section *B* of Fig. 3. By the same process of reasoning, why should anyone manufacturing oils of viscosity indices less than 100 agree to render obsolete oils falling in section *A* of Fig. 2?

The 10-W and 20-W oils are considered by the authors strictly as winter-type oils. Evidence is presented to show that by the use of these oils the motoring public really comes out ahead in so far as oil cost and engine-repair bill is concerned. Certainly, before new classification of winter motor-oil can be adopted, pour-point limits or, preferably, pumpability limits, must be made a part of the classification since it is obviously more important that the motor oil circulate on starting than that the engine start. If the purpose of the 10-W and 20-W grades is to guarantee to the motorist that he can start his car at —15 or 0 deg. fahr. respectively, it should likewise be the purpose of a satisfactory designation to guarantee that the oil would circulate after the engine has started.

A review of the entire subject of winter oil classification leads to but one conclusion; that is, if any revision whatever is necessary in the present S.A.E.-10 and S.A.E.-20 classifications, that the revisions made should accomplish the following results:

(1) That no oil be rendered obsolete

(2) That there should be no overlapping classifications

(3) That if oils are designated as winter oils in regard to their cranking properties they should be likewise designated in regard to their pumpability characteristics, which at present is more or less defined by pour point

(4) That the oils designated as winter oils should prove acceptable to the motor-car industry and to manufacturers and marketers of oils of both low and high-viscosity indices

(5) That, for commercial reasons, the present classifications of oils should be altered as little as possible in attaining the above requirements.

Diesel-Fuel Specifications

SPECIFICATIONS are a very restrictive and bothersome handicap, while being at the same time indispensable as a tool for effecting desired results when properly understood and suitably applied. Certainly the use of ignition-delay or critical compression ratio information should be an invaluable tool to the refiner and the designer in cooperating on the development of the Diesel as a practical prime mover for automotive equipment. Without it, progress will be in essentially a hit-and-miss fashion, without direction and without adequate results.

The one condition which is to the interest of everyone to avoid, especially the engine user, is the use of a working tool as a restrictive, and constrictive check on progress. To use this property of Diesel fuels as a means of forcing the use of a more costly fuel, instead of using it to develop the more efficient use of a cheaper or more abundant fuel, will be a case of the tail wagging the dog—to the painful disadvantage of both. The greatest value that can come from the use of ignition quality data is through use as a research tool—not as a weapon.

Briefly, the job of the technologist is to find out and measure the properties of available fuels in such manner that the information may be of the maximum benefit to the engine designer; to learn how to operate refining units to produce the most economical fuels at the lowest price and in the greatest quantity.

The engine designer's problem is to design to utilize the most economical and most abundant fuel possible for the refiner to produce at the lowest cost; to adapt his product to take advantage of, rather than be handicapped by, the properties of the most economical fuels, to widen rather than restrict the choice of fuels available to the consumer.

The problem of both is to cooperate, each to solve that part of the common problem which he can do easiest and at the lowest cost, to avoid indulgence in the Old Army Game—Passing the Buck.

The situation is reversed with the Diesel fuel problem. In demanding a fuel of high cetene number, the demand runs exactly counter to economic trends.

—Excerpts from a 1934 Semi-Annual Meeting paper on "Prospects for Future Diesel Fuels, and Their Available Supply" by Arch L. Foster, National Petroleum Publishing Company.

Dr. Abbott's Paper on Sound Measurements in Automotive-Noise Reduction Discussed

THE paper was published in the August, 1934, issue of the S.A.E. JOURNAL, pages 271 to 287. It expressed the desirability of measuring sound by soundmeter rather than by listening with the human ear, since soundmeter measurements indicate definitely just what components of noises must be reduced and also just what has been accomplished by any given change.

In the following discussion of the paper resonance noises are cited as a cause of complaints, a direct method of sound analysis is recommended, and emphasis is placed on having acoustical data interpreted by experienced observers.

It is stated and enlarged upon that the two aspects of automotive-noise reduction—the use of acoustical aids to locate weaknesses in manufacture and to locate sounds which annoy the customer—require separate consideration. Noise-elimination materials are cited and commented upon, riding-comfort and noise-problem treatment is analyzed, the time element is emphasized as an important factor of sound analysis, and the term "audibel" is suggested as a substitute for the term "weighted sound-pressure level."

Resonance Noises Cause Complaints

—C. E. Nelson
Burgess Laboratories

I WISH to congratulate Dr. Abbott on his clear, straightforward paper. So far as I know, this is the first attempt to explain in a simple way the troubles encountered in sound reduction and to define some of the involved names pertaining to sound. At times I am led to think that some of these involved definitions were meant to confuse everyone, including the authors of the definitions!

With respect to automobile noises, I believe that resonances are of more importance than Dr. Abbott has pointed out. The customer will put up with quite a bit of unpitched noise; but, as soon as he hears a steady note which he can distinguish from the background noise, he centers his attention on it and the car immediately becomes "noisy," even though the total noise may not be appreciably louder. This is especially true if the note is heard in a narrow speed-range only. Since resonances tend to produce such a condition, they become important items in automobile quieting.

A set of tires or a muffler, or such similar equipment, may have a comparatively high noise-level over the whole speed-range and still be acceptable in many cases, provided that there is no sudden change of noise with speed. If there is a period noise, however, this equipment is rejected in favor of the consistently noisier equipment, but without the "period."

These remarks are not intended to belittle the attempts to reduce these important period-noises, but they are intended to be a reminder that background or unpitched noises are still noises and that it is essential to reduce them also if effective quieting is to be accomplished.

Direct Method of Sound Analysis Recommended

—E. G. Gunn
Walker Mfg. Co.

I AGREE with Dr. Abbott that the measurement of sound as to intensity and frequency is essential in the work of noise reduction, in fact, as useful as a dynamometer in improving engine power. Our work for several years has been largely on exhaust systems, and we prefer what we call the direct method on sound analysis. This method consists of making a polar pressure-wave diagram by the use of a manograph rotated by the engine, and an optical system to trace the diagram on a screen.

Noise coming out of the tail pipe can be divided into two groups:

(1) Notes with a frequency in the order of, roughly, 100 cycles per sec.

(2) Unpitched noises of several thousand cycles per second and up

We have found it next to impossible to make useful diagrams of the high-pitch noises and so have resorted to mechanical filters for studying these noises. Fortunately, however, in the exhaust system, these high-pitch noises yield to simple treatment; so, we are concerned principally with notes.

If we put an adjustable Helmholtz resonator near the end of the tail pipe and connect this to the manograph, the

resonator can be adjusted to show the curve of the critical notes. Thus, with the engine running, changes can be made, and the effect on this one frequency noted on the screen and also by ear. In all cases observed there are several notes to be reckoned with, and this method provides a quick means of studying the effect of changes, on each note graphically and separately. However, determining note frequencies and amplitudes is by far the easiest part of the problem.

For instance, these several notes can be tuned out in the pipe from the end of the tail pipe to the manograph by the use of Quincke filters so they cannot be heard with a stethoscope in the pipe, and so they do not show on the diagram; but, so far, the use of Quincke filters in the exhaust line has failed to accomplish the same result.

Measurement Data Require Experienced Interpreters

—R. F. Norris
Burgess Laboratories

I SHOULD like to compliment Dr. Abbott on bringing to us a paper which so clearly states the problem of sound measurement from the acoustical engineer's standpoint. I thoroughly agree with everything he has said and I think that, because of his paper, we will all have more definite appreciation of the measurement of sound by instrumental means. He has very concisely stated the limitations and the uses of measurements of this kind.

We have been using instrumental methods of sound measurement at the Burgess Laboratories for the last eight years and, in fact, have developed apparatus particularly designed to make sound measurements and sound analyses. I would like to emphasize Dr. Abbott's statements in regard to the absolute necessity for having experienced acoustical engineers interpret the results of the measurements, as well as having them set up the conditions under which the measurements are made. An instrument, however well designed, cannot be substituted for a trained brain.

A factor which makes the acoustical engineer's work more difficult is that the automotive engineer has had so many other problems on his mind that he has not spent a great deal of time in acquainting himself with acoustical terms and procedures and, consequently, must be instructed in these matters by the acoustical engineer's report. I feel that we should spend enough time on the study of acoustical subjects so that we would become thoroughly acquainted with acoustical terms, their meaning and significance. If we do this we can then state our noise problems to the acoustical engineer in definite terms, rather than in the indefinite and unscientific terms which we are prone to use.

Suppose that an automobile manufacturer went to an artist and said: "I don't like my car. Something is wrong with it. I think it's the color and I would like to have it fixed up." The artist paints the car green. The manufacturer says: "Well, that's better; but I don't like it." The artist then paints the car red. The manufacturer then says: "Well, that's better than the other; but still I am not satisfied." The artist paints the car blue and the manufacturer says: "That's worse than it was in the first place." Finally, after many attempts, the artist paints it a canary yellow. The manufacturer says: "Now that's exactly what I want except that it's a little too bright." The artist tones the yellow down and the manufacturer is satisfied. But how much more simple it would have been if the manufacturer had known enough about color in the first place to say: "I would like the car painted yellow and a little darker than a canary yellow."

That is about the plight in which the acoustical engineer finds himself when he is asked to quiet a car. The engineer or manufacturer says: "This car is noisy and I want it quieter," but he has no particular idea as to the frequency range of the noise which he dislikes and is unable to set a standard for the quietness that he desires. All that he knows is that it is noisy and he wants it quiet. It is then the acoustical engineer's job to try and find out by indirect methods what particular noise is disturbing the manufacturer and how to eliminate the noise. The problem of eliminating the noise is sometimes much less difficult than that of determining which noise is objectionable to the manufacturer.

In going through the plants of different manufacturers, one finds any number of standards for quietness. Taking an hypothetical car, for instance, which is noisy in general and has several bad peaks at discrete frequencies, let us assume 120 cycles and 350 cycles for the peaks. If the acoustical engineer worked on this car and reduced the general or background noise by 10 decibels, the peaks would be apparently amplified and undoubtedly the manufacturer would say: "The car is much noisier than it was in the first place." If, however, the acoustical engineer works on the peaks alone and reduces them to the level of the background, the manufacturer will say that the car has been quieted materially. In each case the quieting may have been exactly the same so far as the noise measurements are concerned. If, in the first place, the manufacturer had known enough about sound problems to realize that the peaks existed and asked the acoustical engineer to try to iron out these peaks, the result would have been accomplished in a much shorter time and at less expense to the manufacturer.

To sum up, it does not seem that we are being entirely fair to the acoustical engineer unless we are willing to learn his language to such an extent that we can intelligently state the problem that we would like to have him solve for us and, when he finds the solution, to understand what he is talking about when he makes his report.

Separation of Two Aspects of Automotive-Noise Reduction

—Donald A. Laird
Psychological Laboratory, Colgate University

TWO aspects of automotive-noise reduction should be clearly separated. On the one hand there is the use of acoustical aids *to locate weaknesses in manufacture.* Measurements of this cannot be on too minute a scale, for a difference of a decibel in intensity, which cannot be distinguished by the human ear, may be of tremendous significance in revealing an inferior mechanical condition.

The other aspect is *to locate sounds which annoy the customer.* This side of the work involves not only the intensity of sounds, but their quality. Some pitches are more annoying than others; warbling and intermittency have also to be considered. A relatively faint sound of one type is more distasteful than a louder sound of another frequency pattern. From

the viewpoint of sales, it is the disturbing sounds that should be run down rather than just a general reduction in loudness. There are a few other rather basic points which are worth emphasizing, although they may not involve anything new to the group.

What audiometric instruments record depends upon the surroundings in which the measurements are made. A vacuum cleaner in a corner of a room yields a different acoustical spectrum than is obtained from the same cleaner in the middle of the room. In the middle of one room it will also yield a different spectrum than that obtained in another room. Extremely uniform surroundings are more essential in sound determinations than, perhaps, in any other field of mensuration. One series of measurements could always be taken in a test room which gives the maximum reverberation; this will, in a sense, magnify the readings and show up the parts at their worst. For example, the horn of an automobile always sounds louder in an enclosed garage than out on the open road, and the car is noisier going through a rock cut than on the open road.

Not a great deal can be done to decrease noise as it strikes the passenger by enclosing the chief noise-sources, unless considerable added weight is incorporated in the housing. A soundproof door with a keyhole in it, for instance, is no longer soundproof; almost as much noise will come in through the keyhole opening as would pass through an ordinary panel door.

Noises cannot be subtracted, the way total weight can be reduced. When there are ten typewriters in a room, and one of them is stopped, noise is not reduced by one-tenth. It is, in fact, not reduced at all to the human ear. The major sources of noise have to be quieted first before any appreciable difference will be noted from counterbalancing the generator armature.

The transmission, or telegraphing of noises, offers many problems of practical importance, especially since resonant surfaces or cavities may build up the weak primary source. More consideration should be given, also, to the reduction in noise by the use of unlike materials at points of contact.

I hope that these few brief points will generate some practical discussion. Probably the most important phase is the detection and reduction of noises that give the consumer an unfavorable impression, for there are other methods than acoustical measurements for determining machining accuracy and smoothness of operation, methods which are often more accurate and convenient than audiometric determinations. Unfortunately, however, the reaction of the average person to slight auditory differences is a complicated thing to determine. Work in the last few years by several investigators indicates that it can be done.

Noise-Elimination Materials Divided Into Four Groups

—John S. Parkinson
Johns-Manville Research Laboratories

DR. ABBOTT has covered the subject of sound measurements so thoroughly that there is very little which can be added. I believe that in his effort to attain scientific impartiality he has painted a little darker picture of the difficulties besetting noise measurements than is quite necessary. There is no question but that such problems as microphone placement in a complex wave-pattern and the relative importance of individual frequencies in a complex note present unexpected hazards. Of the whole, however, our experience in measuring automobile noises, as well as other types of sounds, has amply justified the use of various types of sound-measuring equipment. There is no doubt that modern electrical methods of measuring and analyzing sounds present data which are far more accurate and more detailed than the ear can obtain. It is true that a certain amount of experience is necessary and desirable in obtaining and interpreting these measurements; but as Dr. Abbott himself has so amply proved, a capable and experienced observer will find such measurements an invaluable tool in noise reduction.

I should like to call attention to the excellent work performed by the Committee of the American Standards Association on acoustical measurements and terminology. This Committee has a Subcommittee on noise measurements whose chairman is Dr. Harvey Fletcher, director of Acoustical Research of the Bell Telephone Laboratories. This Committee has issued several preliminary reports dealing with proposed standards. The status of this work is described in the Industrial Standardization and Commercial Standards Monthly for May, 1934.

Among other things the Committee has suggested that the term "loudness level" be applied to measurements made in decibels on sound-pressure or sound-intensity levels such as Dr. Abbott has described. It is suggested that the term "loudness" be reserved for a quantity which more nearly represents comparative or relative loudness as judged by the average ear. The Committee's recommendations contain a tentative scale for converting loudness levels to loudness. The recommendation for this scale derives from a fact which Dr. Abbott mentions; namely, that differences in decibel levels do not give the percentage change in the apparent loudness of a sound. This is an inherent difficulty with the decibel scale and it is felt that, by providing a means of conversion whereby decibel levels may be expressed more nearly in terms of human experience, a definite step forward has been taken.

For example, a reduction in noise level from 60 db. to 50 db., when considered merely as a subtracted difference, conveys little or nothing to the untutored observer. However, by referring to the conversion chart mentioned, it will be seen that such a reduction presents approximately a 50-per cent change in loudness for most sounds. Similarly, a 5-db. reduction would have presented a 33-per cent change in loudness, and so on.

Dr. Abbott deals briefly with the various types of materials available for the elimination of noise and vibration. Manufacturers of these materials have recognized the need for more accurate data as to their characteristics, and our own company has been particularly active in studying the development and application of such materials, which can be divided into four distinct groups, as follows:

(1) Sound-Insulating Material
(2) Vibration-Damping Material
(3) Sound-Absorbing Material
(4) Vibration-Insulating Material.

Each of these products has a special function and possesses special characteristics of its own. The knowledge of these characteristics is an essential to their intelligent use.

A sound-insulating material is intended to stop the transmission of sound from one locality to another. Its principal

requirement is reflectivity and, consequently, it must possess stiffness and mass. As Dr. Abbott has pointed out, glass and steel are good sound-insulators. Porosity is not a requirement, and may indeed be a detriment. An insulating material may well be finished with a soft or porous surface to lessen radiation, but it must possess the other physical requirements first.

A sound-absorbent material, on the other hand, is usually porous or flexible. The efficiency of a sound-absorbent material may usually be judged by its density, permeability and by the porous character of its surface. Its applicability to a special problem should be judged by its absorption-frequency characteristic; the range of frequencies at which it is efficient. This characteristic is readily measured and no material should be sold without such data.

A vibration-damping material is intended to prevent the vibration of a solid membrane such as a body panel. There are a number of such materials on the market, many of which really fall into the classification of sound-absorbent material rather than vibration dampers. The important characteristics of a vibration-damping material are high internal resistance and, to a lesser extent, mass. Such a material may well be combined with a sound-absorbent or sound-insulating construction; but the separate functions should be carefully kept in mind.

The vibration-insulating material is intended to prevent the transmission of vibration from one part to another. Flexible mountings under the engine or rubber pads between the frame and the body serve this purpose. The important requirement for such a material is that it have a high compliance, that is, that the deformation for unit load be great. This characteristic should be permanent for, if the material alter in its physical characteristics with age, it is obviously unreliable. In some cases a fairly high internal resistance is desirable for a vibration-insulating material, but this will depend upon special circumstances. Neither mass nor porosity are of any value by themselves.

I have sketched this brief outline of materials because I feel that it is of the utmost importance that the automotive engineers have a clear conception of what they are buying when they seek to achieve noise reduction by such methods. Too often a material is described in vague and general terms which makes it seem a panacea for all types of problems. To date, no such material has even been found and each specific problem requires its own solution.

Riding-Comfort and Noise-Problem Treatment Studied

—C. A. Tea
Chrysler Corp.

DR. ABBOTT has made a valuable contribution to the relatively new science of noise study in explaining the theory of sound measurement, instrumentation and interpretation of sound measurements. His presentation of the facts governing all sound studies is very timely and should be of great value to the automotive engineer engaged in noise reduction work.

It is true that noise meters are only as valuable in noise work as their records of decibel measurements are properly interpreted and they are often very unreliable where they depart from the well-known Kingsbury curves. For investigating sounds in narrow bands, the noise meter is an indispensable tool and cannot be ignored if one is to reduce noise efficiently. In regard to analyzers for general automotive work, we have had some very valuable experience with some of the best-known makes and have found that the band-pass-filter type with bands varying in octaves from 64 cycles up is most satisfactory. Such an analyzer does not have too sharp a tuning characteristic and the bands are so graduated as to be of maximum assistance in locating predominant frequencies in the lower end of the spectrum where noise is loudest.

Problems Confronting the Automotive Engineer

The General Problem of Riding Comfort.—The problem of reducing noise in the automobile is invariably associated with improving the riding qualities, as both deal with the ultimate effect on passenger comfort. This, of course, involves not only direct stimulation but also fatigue, usually not realized on short trips. For example, a car that on a 25-mile ride is adjudged a comfortable car because the passenger is not subject to excessive accelerations might be very uncomfortable on a 200-mile trip due to excessive noise, and vice versa. There are other factors that play their roles, too, in the riding-comfort problem, such as eyestrain, although they are of much less importance in the aggregate.

Noise and riding discomfort, in the narrow sense, are thus inevitably involved in the havoc played by vibratory forces on the human mechanism. In one case these consist of the high-frequency forces in the auditory range; in the other, those in the feeling range. Just as sound energy or, as Dr. Abbott puts it, "sound pressure," when not regulated, can produce nervous discomfort and headaches, so also excessive mechanical vibration of visual proportions causes dizziness and actual physical pain.

We have developed a precision riding-quality measuring-instrument that, like a noise meter, gives in one reading the condition of a car over any road at any specific speed above a predetermined level of zero physical disturbance. This reading represents the integration of all maximum forces or accelerations in proportion to their values received by the passenger and their durations regardless of the quality of the vehicle's vibration, and is truly representative of the ride of a car at any particular speed or at any location in the car. This instrument, which gives a true riding-quality factor and is sensitive enough to distinguish between a few pounds of tire pressure, will be discussed at length some time in the future.

Attacking the Noise Problem

The modern automobile consists essentially of a closed compartment surrounded by numerous sources of ever-present variable-noise. To mention only a few, the principal ones include the engine, fan, muffler, carburetor, gears and road wheels. Since these send out waves of different mixed frequencies and intensities, depending upon the speed and torque of the automobile, the problem of reducing their individual loudness and subsequent transmission to the body by air, resonant energy or direct transmission, is often very complex because they cannot be readily separated and studied as individual noises. Usually, a careful frequency analysis of the whole will indicate what parts are producing the greatest noise by a knowledge of their individual functions and characteristics.

Assuming that there are no engine, no transmission and no accessory parts and that the car is moving at high speed, con-

siderable noise will enter the body due to a resonant condition caused by vibration of the tires set up by road impact. This initial vibration is of the order of 60, 120, and the like, cycles per sec.; also, a single shock produced by impact can result in body noise by originating vibration in such susceptible parts as frame, body and fenders. To reduce this source of noise, which is probably the greatest contribution today to nervous fatigue, the body should be isolated as far as possible by rubber and other known vibration-absorbing materials. The stiffening up of freely vibrating parts such as large unsupported panels will add greatly to the diminution of noise by partial elimination, at least, of resonance effects and for another good reason to be mentioned later. Strangely enough, varying the tire pressures in small increments above or below normal adds to rather than subtracts from the general noise. Wind noise in correctly streamlined bodies is unimportant.

If we assume that the engine and all the component parts are replaced in the car, complications immediately arise and sound instruments become indispensable in the work of sorting out individual sources of sound for reduction and gaining an harmonious balance. Usually, with good engine mountings and balanced engine, the relative importance of this part is greatly diminished and such peak-sounds as are produced by the muffler and fan stand out prominently in the noise picture. Both muffler and fan periods can be determined with a good analyzer. The fan fundamental-frequency is usually approximately equal to the speed times the number of blades and, since both pitch and loudness increase with the speed of the engine, it is not such a simple matter to quiet this part.

Our experience indicates the importance of attempting to balance frequencies by partial reduction and adjustment of noise intensities to effect a spectrum of uniform ear-response. For example, low "drumming" noises when raised in frequency or reduced in intensity tend to produce much more apparent quiet than reducing or even eliminating higher-frequency notes. Where ordinary noise-meters fall short is in their inability to recognize frequency and this, even more than total loudness, influences the pleasantness of mixed sound which is usually very difficult to reduce appreciably.

Body Treatment

After a complete sound-analysis is made in a car, those frequency bands most prominent (usually low) should be worked on to reduce the intensity within desirable limits by attacking the noise at the source. What noise is left must be steered clear of the body by adequate absorption, insulation against air-borne noise and mechanical damping of resonant parts. Since body noise originating at the road wheels is of very low frequency, between 64 and 256 cycles per sec., raising the natural frequency of body panels (floor, walls and ceiling), often will bring about improved results for two reasons; namely, by reducing resonance and, indirectly, by permitting absorbent material to be used to its greatest advantage.

The relative decibel reductions effected by bracing large unsupported panels and applying "deadening" or sound-absorbent material in a particular car under test are illustrated in Fig. 1. The greatest noise-reduction was achieved by stiffening up certain floor panels without the aid of absorbing material as indicated by Curve 2. A vibrometer is a convenient tool for judging panel vibration, while a noise meter must answer for the final results of treatment. Curve 3 shows the extra reduction gained by the addition of a very efficient damping material. All insulating materials absorb higher frequencies better than low; in fact, tests show that, for sounds under approximately 256 cycles per sec., no pure sound-absorbing material is worth its weight. Fig. 2 shows absorption curves of two well-known commercial materials used in automobile bodies.

Finally, for achieving refinements in noise reduction, good sound-absorbing material should be used intelligently and placed where it will do the most good or not used at all. Damping of body panels is sometimes important to combat resonance, but it should be done with attention paid to the effect on the fundamental or harmonic frequency or pitch of the

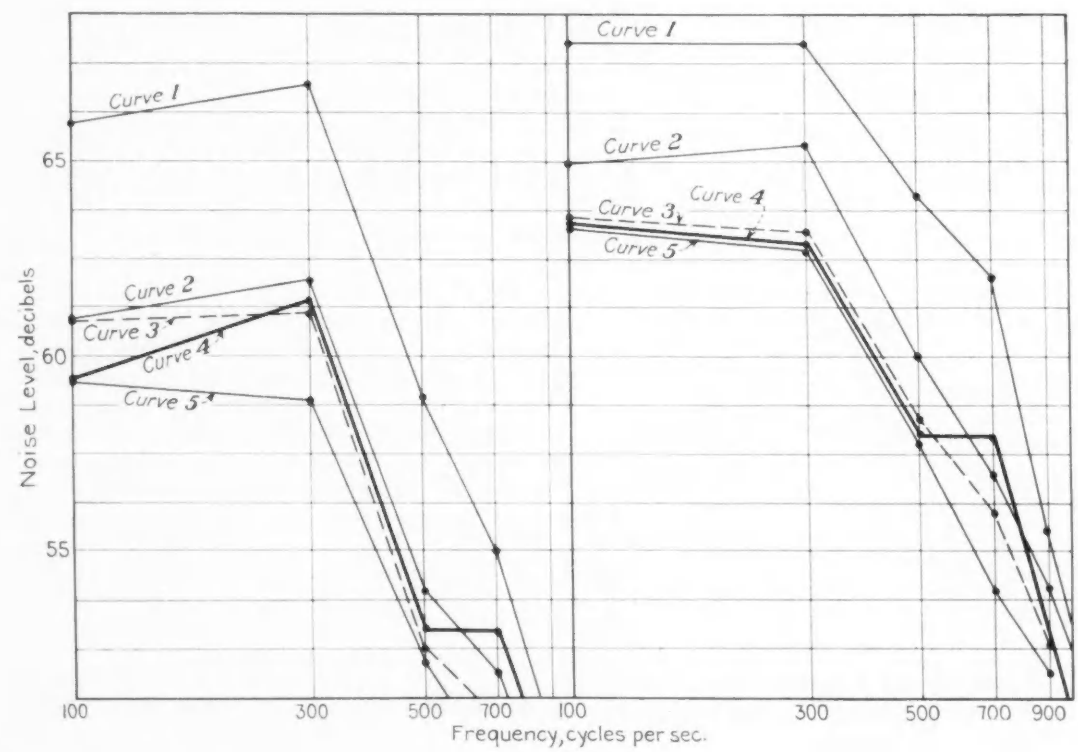


Fig. 1—Noise Analysis of a Four-Door Sedan

The meter was located in the rear of the car at car level. Curve 1 was obtained before treatment; Curve 2, after bracing the large floor-panels; and Curve 3, after treating the large floor-panels with efficient damping material. The curves at the left were obtained at a speed of 30 m.p.h.; those at the right, at 40 m.p.h.

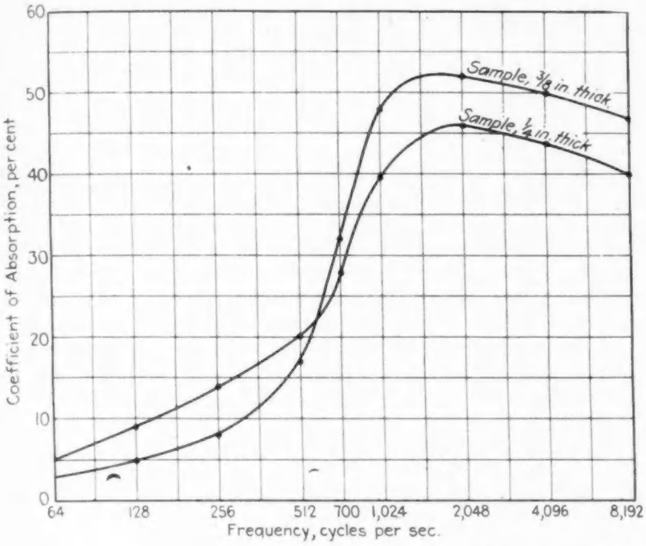


Fig. 2—Sound-Absorption Curves of Two Commercial Materials

part to be "deadened." In some cases, the second or third harmonic of a panel dictates the pitch just as in a bell and these, unfortunately, are not capable of mathematical derivation. Some "deadening" materials are really disadvantageous because they lower the tone and simply bring the part into resonance, with more intense vibrations resulting.

In conclusion, we have only scratched the surface in dealing with the noise problem in an automobile, and much will be left to the instrument makers and to the development of technique before great strides can be made in formulating definite methods for quieting motor vehicles.

Time Element an Important Factor of Sound Analysis

—G. T. Stanton

Electrical Research Products, Inc.

DR. ABBOTT is indeed to be congratulated on the completeness of his paper, summarizing many of the factors involved in the measurement and analysis of sound. The complexities and difficulties of the problem are quite completely stated as well as the outlining of certain fundamental conceptions, and possible misconceptions, of the meaning of noise measurement.

I should like to add one point to the fundamental consideration. A complete statement of the analysis of sound in factor terms requires, in addition to intensity and frequency of the components, a consideration of the element of time. This element applies in two ways; one, the actual duration of the elements and, in many types of sound, the relative time of occurrence of the various components. This time pattern is of a material degree of assistance to the ear in separating the various parts of complex noise. Thus, an engine mechanic, in listening to the noise of an engine, distinguishes between piston-pin knock, piston slap and tappet noise, partly by the characteristic frequency of each sound and their relative intensities and, to a large degree, by the timing order of their occurrence.

Perhaps some listeners not practically acquainted with the applications of sound measurements to noise reduction may feel that the existence of the tremendous complexities pointed out in this paper present a somewhat insuperable barrier to the immediate practical use of this valuable tool. While I can testify at first hand to the existence of the many complexities, it is believed that, for practical use, many of these difficulties can be removed.

Noise, as produced by most machinery, is complex. The usual physical distinction between musical sound and noise defines a fundamental with a series of harmonically related overtones as a "musical sound," and "noise" as a series of inharmonically related components. While basically true, this definition implies a simplicity which exists only in rare circumstances. Fixed-speed rotating machines may produce a limited number of sound components consisting of fairly constant intensity and frequency. In addition, however, there will be an almost infinite series of components generally called "unpitched" sounds. This latter sound will generally be found to be made up of components, the individual intensities of which fluctuate through wide ranges. At one moment they may be non-existent and, the next, of high intensity, some groups occurring in a regular time succession, others in a completely random fashion.

This is illustrated by Figs. 3 and 4. Fig. 3 represents a sound, artificially produced, consisting of an extended series of individual components representing the classical conception. This record was made by sweeping a narrow band-pass filter through the frequency spectrum, the intensities being recorded by a high-speed continuous recorder. Fig. 4 is representative of the noise produced by a high-speed electric-drill, recorded in the same way. Definite components may be seen to exist at certain frequencies, but the majority of the energy may be seen to be almost continuously distributed, some energy existing at nearly every frequency. This is representative of most types of rotating-machinery noise.

One of the early methods employed for analysis, and an obvious one in view of the conception of noise as a series of individual components, was the use of some form of resonating device. For convenience, these resonators were generally electrical or mechanical and their resonance frequency adjustable. For the laboratory case consisting of a finite number of individual components, approximately fixed in magnitude and frequency, this device proved quite useful.

In the special case of fixed-frequency rotating-machinery, certain components directly associated with the speed of rotation could be determined. Inasmuch, however, as the majority of noises from most machines consists largely of components fluctuating rapidly in both magnitude and frequency and containing in many instances large quantities of so-called "unpitched" noise, the use of this form of analyzer became impracticable. It was obviously not feasible to secure measurements of the intensity at each and every frequency throughout the spectrum and, if only a limited number of large-magnitude components were reported, an inadequate picture of the true conditions resulted. In addition, the duration or time sequence of energy at different portions of the spectrum could not be determined or reported in this manner.

Electrical filters arranged to divide the frequency spectrum into several parts were found well adapted to the study of noise in terms of its annoyance or physiological effect; but, unless a prohibitive number of filters was employed, the detail was far too limited for an engineering study of the causes of noise.

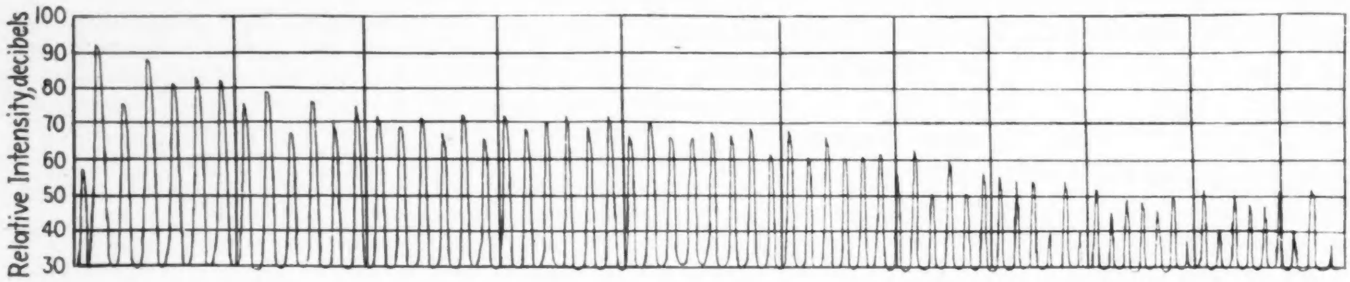


Fig. 3—Illustration of Sound, Artificially Produced, Consisting of an Extended Series of Individual Components Representing the Classical Conception. A 20-Cycle-Band Tone; 360 Db. per Sec.

An analyzer has recently been developed containing the desirable features of both the resonant and fixed-band filters and, in addition, the ability to listen to and identify by time-pattern the energy in any portion of the spectrum. In general, two widths of pass-band, one of 200 cycles and one of 20 cycles, are available. Both of these bands may be swept through the spectrum continuously or adjusted to pass any desired band of frequencies. Where it is desired to isolate and measure individual components, the 20-cycle band is of sufficient discrimination for all but the most unusual cases, and is in fact as narrow as usual frequency fluctuations in rotating machinery will permit, due to continuous slight speed changes. Extremely high suppression of energy outside the band and constancy of frequency calibration is obtained by an unique use of the piezo-electric crystal filter.

Instead of the indicating meter on the analyzer, generally used for most analyses, the input of this meter may be associated with a high-speed recording-meter in two different ways. The pass-band of the analyzer may be swept rapidly through the spectrum synchronously with the recorder chart, thus giving a complete intensity frequency spectrum in one brief operation. The charts of Figs. 3 and 4 were so made. For other uses, the output from a single band of frequencies is passed to the recorder and the time-intensity fluctuation of this particular band may be recorded for further study. The first method is particularly useful where the sound in question cannot be prolonged a sufficient period of time for a step-by-step analysis, or where a large quantity of units are to be studied in rapid succession. The second method is of particular value for the evaluation of irregular or intermittent types of sound.

The pass-band analyzer materially reduces the inherent difficulties due to standing-wave, minor-frequency fluctuations and the like, that are present when measurements are made of single finite frequencies and, generally, values sufficiently representative for engineering work may be obtained by averaging a comparatively small number of readings. In addition, the time element required for the making of a spectrum analysis is materially reduced. By combining the ability to listen to the portion of the spectrum in question simultaneously with its measuring, considerable aid to rapidity and accuracy of determination is offered.

Dr. Abbott has summarized seven major classifications of noise reduction which cover quite generally the methods available. I should like to emphasize the importance of accurate measurements of the effect obtained by the application of any one of these methods in a determination of the choice of method. It is frequently possible to combine in the application of a material, or in the change of design of a part, not one but a combination of several of the above methods. Obviously, the best and most economical solution can be attained when the effect of such addition or alteration may be independently determined for each of the factors.

For instance, the three items of enclosure, acoustical absorption and introduction of mechanical damping-resistance may each be affected in a desirable direction by the judicious use of a single material. In the selection of this material and of its method of application thought should be given to its effectiveness for each of these three factors, that method being selected which gives the maximum overall improvement.

While time does not permit covering the manifold applications of the quieting of an automobile which have been made with modern noise measurements, I should like to select one of the five items mentioned by Dr. Abbott, that of the introduction of mechanical damping-resistance, which he mentions largely as a theoretical application.

A recent instance occurred which brings out rather clearly the value of instrumentation in determining the cause of and reducing a rather serious noise-problem. A disturbing road rumble was present in a recent model of all-metal body-construction. This rumble took the form of a low-pitched "grunt" in passing over strips in a concrete road, becoming an almost continuous "roar" on gravel, brick or other rough-surfaced roads. Damping material of various sorts had been applied without noticeable results until instrumentation was resorted to. An analysis of the distribution of frequency of car noise at a variety of speeds on varying road surfaces indicated that the energy was concentrated in a relatively narrow

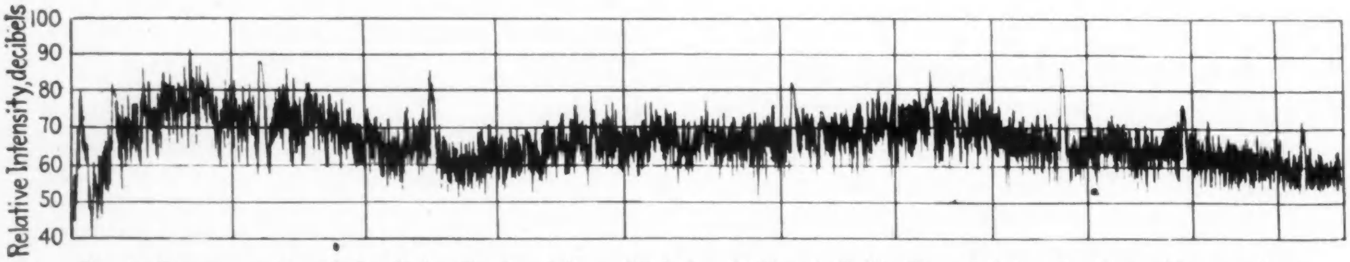


Fig. 4—Representation of the Noise Produced by a High-Speed Electric-Drill, Recorded in the Same Way as for Fig. 3. A 20-Cycle-Band Noise; 360 Db. per Sec.

band of low frequencies. By adjusting the pass-band of the analyzer to cover this particular frequency, it was possible to determine rapidly the effect of even small changes. The predominant frequency in the car noise agreed very closely with that produced by pounding on the frame with a rubber hammer.

It was thus indicated that the noise was in the form of a transient at the natural period of the body panels. The duration of the sound was fairly short and, consequently, we expected a material increase in apparent loudness as the duration of the sound was prolonged. It was believed, therefore, that the addition of mechanical damping would materially reduce the duration of these sounds, as well as reduce their peak intensity. By locating the microphone in the vicinity of the suspected panels, the effect of application of damping was clearly distinguished by measurement, although the general overall noise was not materially changed.

Here is where cut-and-try methods failed since, in previous attempts, as the overall car-noise was not reduced by the treatment of a single panel, the method had been rejected. By continuing this method step by step until the last seriously offending panel was located and damped, an improvement of about 8 db. in loudness of the rumble was accomplished and a total reduction in overall car-noise of about 4 db.

At this time a secondary-frequency peak was found and determined by the above methods to be a vibration in the floor board excited by the propeller shaft and differential gears. In this instance Method No. 1 was used, that is, mismatching vibrational impedances, together with strategic addition of damping, the ultimate result being a reduction of approximately 12 db. in rumble noise-level at 40 m.p.h.

In conclusion I should like to point out that, while all of the difficulties and complexities mentioned by Dr. Abbott actually exist and must be thoroughly considered in any attempt to make noise measurements, it is, however, completely practicable, by a thorough understanding of those features which may be neglected in a given problem or by the selection of apparatus to minimize certain effects, to make noise measurements rapidly and with sufficient accuracy to serve as an unerring guide in a determination of the accomplishments of any design changes and to point the way in the selection of those changes which should be made to accomplish further reduction. Mention must also be made of the application of acoustic measurements in a laboratory determination of efficiencies of materials for acoustical absorption, mechanical damping-resistance and the efficiency of transmission reduction of enclosures. In combining a material to add acoustic absorption or mechanical damping, it would appear to be sound engineering to have definite information on the efficiency of the materials intended for these purposes.

The Term "Audibel" Suggested

—W. C. Keys
U. S. Rubber Co.

As Dr. Abbott says, the term "weighted sound-pressure level" is awkward. To replace it, I suggest the word "audibel," which suggests audibility but ends with *bel*, which seems to be necessary for "sound" reasons. It should be noted that acoustical terminology is now being considered by the Acoustical Society.

By-Products of Efficient Streamlining

THE benefits of efficient streamlining are chiefly confined to the lowering of air resistance which accomplishes a great saving in power for any given speed or a great increase in speed for any given power. There are, however, other advantages which are not so well known. Among the most important of these are efficient ventilation (and heating) and added comfort resulting from increased interior space.

That the ventilation problem is still a serious one, despite patented systems now being generally used, is becoming more and more generally recognized. Few engineers realize the fundamental cause of bad ventilation and the bad effects of it on passengers. An important factor is the negative pressure of the air inside the cars and the extent of the area of negative pressure outside the cars. The bad effects on the passengers result chiefly from the extent of carbon monoxide gas prevalent in the passenger compartments.

Because of the extent of the area of negative pressure around a conventional car, the dust and dirt which the car creates or stirs up as it travels cannot escape beyond the immediate proximity of the car and the negative pressure inside the car acts to attract rather than to repel them. Open windows or even front windshields may tend to dispel bad air, but they sometimes tend to increase the amount of carbon monoxide within the car by sucking in from underneath and the sides of the car exhaust gases which would not enter if the windows were closed.

In one test made by Jaray the passenger compartment was found to contain 12.81 cu. in. of carbon monoxide in 35.31 cu. ft. with the side window open and 10.98 cu. in. in 35.31 cu. ft. with the window closed. Only air with as much as 0.05 per cent of carbon monoxide can be fatal, hence the problem of bad ventilation may not extend much beyond the realm of headaches and exhaustion but it is important, nevertheless, and should receive greater attention. In properly streamlined cars a positive pressure can be made to prevail in passenger compartments. In such cars there are points where positive pressure is built up and the area of negative pressure as compared with conventional cars is comparatively uniform and relatively small. By introducing a small opening in the windshield at the point of highest pressure, air is introduced at increased pressure and lowered velocity. This air is pure and since it fills the interior with a positive pressure openings at other points result in expelling used air and may be made to maintain a continuous change without drafts. At various speeds the following positive pressures are maintained in a streamlined car whereas in the conventional car the pressures are negative. . . .

Since in a well streamlined car the fender and running boards are eliminated and width of the body is increased to include the space these parts normally occupy, increased roominess inside the car shell becomes a distinct contribution to comfort. The retention of fenders and running boards is no longer necessary except, perhaps, as a sop to convention.

—Excerpts from a 1934 Semi-Annual Meeting paper on "Streamlining — Up-to-date Facts and Developments," by Lowell H. Brown, Jaray Streamline Corp., and Herbert Chase, Consulting Engineer, New York City.

Slonneger-Fischer Indicator-Diagram Papers Discussed as to Other Phases

TWO papers are commented upon in the following discussion, that by J. C. Slonneger being entitled "Effective Combustion as Determined from the Indicator Diagram" and the one by Hans Fischer bearing the title "Typical Indicator-Diagram Analysis Concerning Effective Combustion." Both papers were published in the August, 1934, S. A. E. JOURNAL, the former beginning on page 288 and the latter on page 291.

Mr. Slonneger's paper set forth a simple and rapid method of obtaining from the indicator diagram data on how, when, and at what pressure combustion takes place in an internal-combustion engine.

The paper by Mr. Fischer pointed out what a desirable indicator card for effective combustion should be, one in which it is indicated that the increase in rate of burning is comparatively small and the rate of pressure rise is very moderate for the high specific output.

Polytropic-Curve Measurement of Effective Combustion

—L. C. Lichty
Yale University

THE papers by Mr. Slonneger and Mr. Fischer are very interesting from several angles, but particularly so from the use of polytropic curves as a means for measuring "effective combustion" in an internal-combustion engine. The method depends on the assumption that equal quantities of energy are required in passing from one polytropic to the other, regardless of the path. No proof is given for this relationship and the accuracy of the assumption might naturally be questioned, since only those properties of a substance that are a function of the state are independent of the path of the process.

At the beginning of combustion which starts at the first polytropic the air has been compressed and contains internal energy, E_1. During the process from the first to the second polytropic, fuel is added which contains internal energy, E_F, flow energy P_FV_F and chemical energy C, of which $1C_2$ has been liberated and C remains unburned. Also heat, Q, is lost to the walls and work, W, is done due to piston movement.

When the second polytropic is reached, the air is partly changed to products, the entire mixture having an internal energy of E_2. The principle of conservation of energy states that the sum of the energy at the beginning of a process plus the various energies added or abstracted must equal the final energy plus the energies removed during the process. Thus:

$$C + E_F + P_FV_F + E_1 = Q + W + E_2 + C$$

or

$$1C_2 + E_F + P_FV_F + E_1 = Q + W + E_2 \qquad (1)$$

Neglecting $E_F + P_FV_F$, which are comparatively small, and eliminating Q by reducing $1C_2$ a like amount and calling it the effective chemical energy, we have:

$$\text{Effective } 1C_2 = W + (E_2 - E_1) \qquad (2)$$

Internal energy, E, does not depend on the path, but work does; so, the effective chemical energy liberated between any two given points on two polytropics, or anywhere else for that matter, does depend on the path followed.

To check the author's assumption, assume a series of polytropics $PV^n = C$, $PV^n = 2C$, $PV^n = 3C$, and the like, to be intersected by a combustion process following a path $PV^{n-\bullet} = K$. or $PV^m = K$ as shown in Fig. 1. It can be shown that:

$$\text{Effective } 1C_2 = \frac{R}{1-m}(T_2 - T_1) + C_r(T_2 - T_1)$$

$$= (T_2 - T_1)\left(\frac{R}{1-m} + C_r\right) \qquad (3)$$

and also that

$$\text{Effective } 2C_3 = (T_3 - T_2)\left(\frac{R}{1-m} + C_r\right) \qquad (4)$$

These relationships neglect the fact that the molecules of mixture are increasing, due both to the introduction of fuel and to the combustion process. Thus, $(1C_2) = (2C_3)$ if $T_2 - T_1 = T_3 - T_2$; or, in other words, the assumption will be true for a polytropic combustion process if equal temperature differences exist between the polytropics for the combustion process.

It can be shown for the foregoing series of polytropics that $\frac{V_1}{V_2} = (2)^{1/(m-n)}$, $\frac{V_1}{V_3} = (3)^{1/(m-n)}$, etc., from which

$$T_2 - T_1 = T_1\left[(2)^{(m-1)/(m-n)} - 1\right] \qquad (5)$$

and

$$T_3 - T_2 = T_1\left[(3)^{(m-1)/(m-n)} - (2)^{(m-1)/(m-n)}\right] \qquad (6)$$

The bracketed terms will be equal when $n = 1$. Also, when $m = \infty$, the combustion process is constant volume and the difference in temperature between the polytropics will be equal. Thus, only for the constant-volume process or if the polytropics were isothermals would the assumption be true. Since the Otto cycle approaches the constant-volume process, the method would appear to apply quite accurately to this cycle except for the error introduced by molecular change.

The combustion path, however, does not follow a path of PV^m = a constant but follows an irregular path as indicated in Fig. 2 of the paper. Transferring the data in Fig. 2 to pressure-volume and also temperature-volume coordinates makes possible the solving of Equation (2). Assuming a temperature of 560 deg. fahr. absolute for the temperature at the beginning of combustion and a compression ratio of 13 to 1, (Fig. 2) of this discussion, the following results were obtained per mol of air:

Point	Temperature, Deg. Fahr. Absolute	E, B.t.u.	ΔE, B.t.u.	Work, B.t.u.	C, B.t.u.	Percentages, Based on Path	Percentages, Author's Assumption
B	1,230	6,160					
			5,420	670	6,090	31.3	31 1/3
M	2,250	11,580					
			4,330	2,260	6,590	33.8	31 1/3
g	3,000	15,910					
			1,850	4,120	5,970	30.6	31 1/3
d	3,300	17,760					
			1,240	2,080	840	4.3	6
E	3,100	16,520					
				Totals	19,490	100.0	100.0

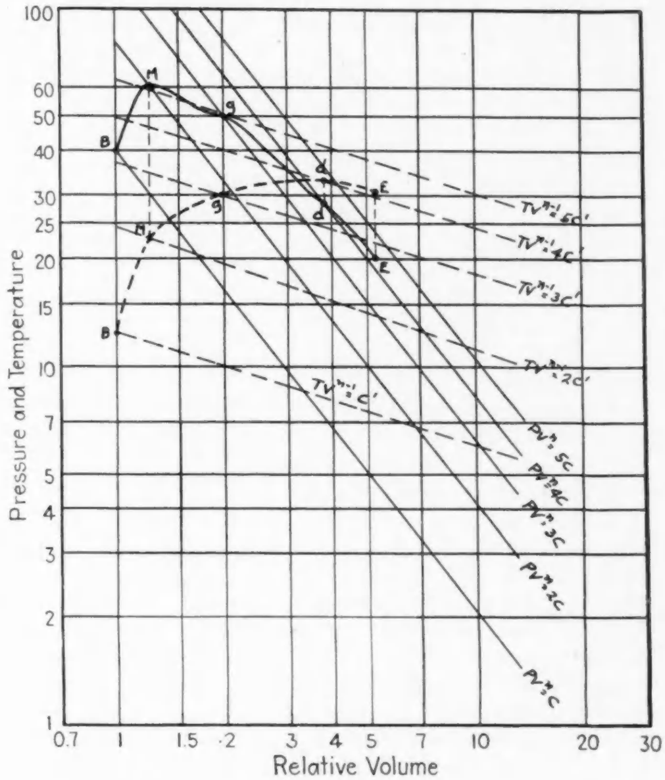


Fig. 2—Logarithmic Graph of the Process on $P - V$ and $T - V$ Coordinates

To obtain the pressure in pounds per square inch, multiply the ordinates by 10. To obtain the temperatures in degrees fahrenheit absolute, multiply the ordinates by 100

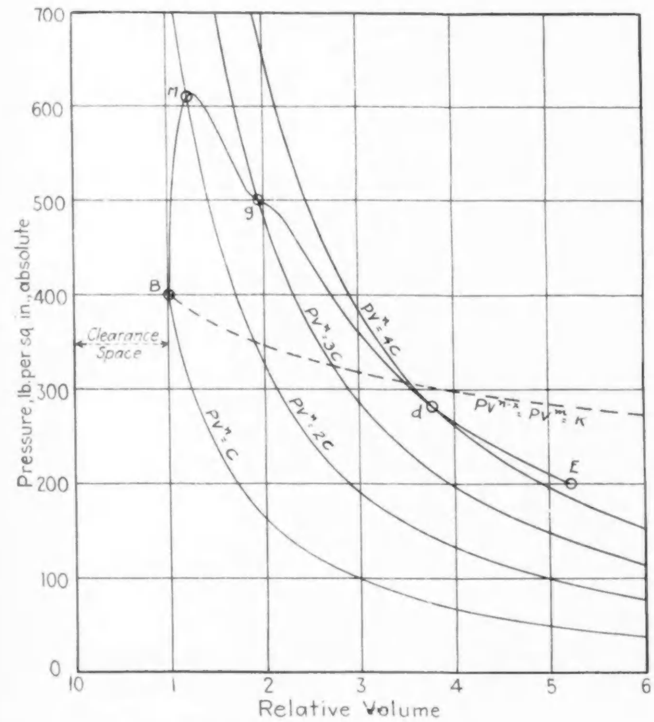


Fig. 1—$P - V$ Diagram of the Process

This shows that, for the process indicated in Fig. 2 of the paper the effective combustion is practically the same between the first and second polytropics, and between the third and fourth polytropics, but is about 8 per cent higher between the second and third polytropics. However, a comparison of the last two columns does not show much difference between the percentages due to disregarding the path followed, at least in this case.

At B, the mixture is all air; but, as the combustion process proceeds, CO_2 and H_2O are formed and less air is found in the mixture. This would raise the energy values, E, at the higher temperatures and increase the difference found between the various polytropics. The heat loss to the cylinder walls probably increases as the higher polytropics are crossed, since the combustion temperature is increasing except from D to E. Thus, the actual chemical energy liberated for the case chosen probably is considerably different between the various polytropics. However, the authors have very skillfully eliminated the heat-loss effect with the definition of the effective-energy liberation. Also, the difference between the assumption of equal effective-energy liberations between the polytropics and what actually occurs is not so questionable when the simplicity of the authors' analysis is considered. The method appears to be a simple and practical way to study the rate or effect of various changes on the rate of burning of fuels in the internal-combustion engine.

Effective Combustion Determined by a Slightly Different Method

—Charles F. Marvin, Jr.
National Bureau of Standards

THE method of following the progress of "effective combustion" on indicator diagrams described by Mr. Slonneger and applied by Mr. Fischer appeals strongly because of its extreme simplicity and because of the very practical value of information which it yields, especially to those seeking to analyze and control combustion in fuel-injection engines.

Indicator diagrams for spark-ignition engines were analyzed some time ago at the National Bureau of Standards by a slightly different method yielding very similar results but permitting a more accurate determination of the point of complete "effective combustion." The method used at the Bureau and results showing percentage of mass burned plotted against crank angle for a variety of diagrams were published in N.A.C.A. Technical Report No. 276, from which the following is a quotation:

"A portion of a typical indicator diagram is shown plotted on logarithmic coordinates in Fig. 3. The solid line shows actual pressures as measured by the indicator and the dotted line represents the pressures which would have been expected had the charge been instantly burned at upper dead-center as in the theoretical Otto cycle. Theoretically and actually, compression is represented on the logarithmic plot by a straight line to the point where the spark occurs. Here the actual diagram begins to rise more sharply due to the burning of the charge. By the time the piston has reached upper dead-center, the actual pressure is at *c* instead of at *b* where it would have been had there been no burning. If it is assumed that the pressure rise above the straight compression-line is proportional to the mass of charge burned, then the fraction of the total charge which is actually burned at upper center would be equal to $(c\text{-}b)/(d\text{-}b)$. In a similar manner, the fraction burned at any other instant during combustion can be computed from a similar ratio of pressures. At *e*, combustion is complete and expansion of the burned gases is represented by a straight line."

The slope of this straight expansion-line may or may not be the same as that for the motoring diagram as is assumed in Mr. Slonneger's method. Due to the presence of products of combustion and the increase in specific heat with temperature, the actual combustion gases have a considerably lower adiabadic exponent than does air at the relatively low temperatures of the motoring diagram, this being particularly so for Otto cycles. Much lower expansion exponents were consistently obtained in a series of diagrams for an aircraft engine in the Altitude Laboratory at the Bureau, as shown in blueprint No. 23.

The slopes of expansion lines on motoring diagrams would presumably be at least as steep and probably steeper than those for the compression lines on the power diagrams. In applying Mr. Slonneger's method to the present diagrams, a family of straight parallel lines (polytropic curves) would be drawn with the same steep slope as the motoring expansion lines. It is obvious that the more gradually sloping expansion lines of the actual diagrams will continue to rise through such a family of lines until the exhaust valve opens, whereas "effective combustion" is actually complete at the point where the expansion line for the power diagrams becomes essentially straight. This source of error in the method doubtless accounts, in part at least, for the apparently prolonged after-burning indicated in Mr. Fischer's Fig. 5.

Indicator-Operation Experiences Related

—E. H. Hamilton
New York University

I WOULD like to state that I have had exactly the same experience with high-speed indicators that both Mr. Fischer and Richard Stansfield of the Anglo Persian Oil Co. have had. We, likewise, purchased one of the British $300 Micro indicators and obtained indicator cards that resembled exactly the type described by Mr. Stansfield.

This particular indicator draws cards 1/10 in. high and ⅛ in. long on a celluloid disc. Obviously, to make any practical use of these tiny cards, they usually have to be magnified 100 times. This magnification also magnifies the lines of the card; hence, when observing the diagrams through a microscope, the diagram looks more or less like a triangular plot of ground bound by a plowed furrow. Any looseness whatever in the mechanism of the indicator would produce an error approximately double the amount of this looseness on the celluloid disc. When this is magnified, obviously, the shape produced by the looseness in the mechanism would constitute a very large proportion of the diagram. This was probably the cause of the jagged diagram to which Mr. Stansfield has referred. The reducing motion furnished with this indicator is theoretically wrong. The motion given to the drum from this reducing motion approaches simple harmonic motion, whereas the drum should be rocked in phase with the piston of the engine which, of course, is far from being simple harmonic motion, due to the relatively short length of connecting rod with respect to the crank arm.

After wasting a lot of time with the Micro indicator, one of the Maihak indicators described by Mr. Fischer was obtained, and, after having used it for the last two years, I find that it is a very reliable, rugged instrument. It requires a minimum of adjustment and can be operated by anyone familiar with the ordinary type of steam-engine indicator. For speeds in excess of 2000 r.p.m., it is considerably affected by the inertia of the moving parts; but, so long as the cards obtained from it are used purely for comparative study of combustion characteristics, this error due to inertia will not actually affect the results to such an extent that they are unusable.

One feature of this indicator not brought out by Mr. Fischer is the reducing motion furnished by the manufacturers. This is one of the few reducing motions which I have seen that gives a theoretically perfect reduction for the indicator-drum motion. I believe that this latter piece of apparatus has as much to do with the obtaining of good indicator cards as the indicator itself.

Heat Required Varies with the Crank Angle

—H. C. Gerrish

National Advisory Committee for Aeronautics

MR. SLONNEGER'S method of indicator-diagram analysis depends upon the assumption that the "heat required to increase the power curve from one polytropic curve to the next is a constant and independent of the path." This assumption is contrary to accepted thermodynamics, which considers that the total heat input is a function of the path. It would be interesting to know just how Mr. Slonneger's method takes into account the heat utilized in performing work.

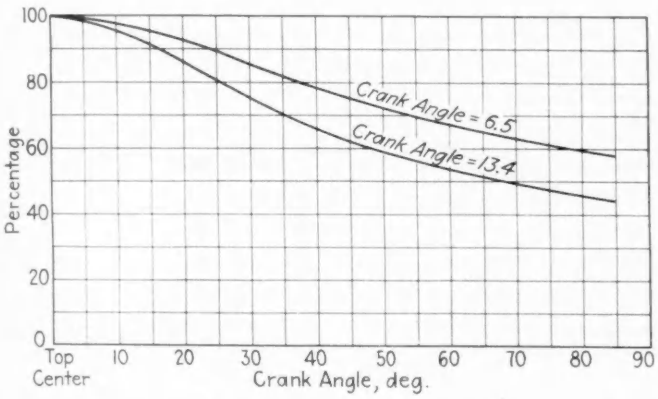


Fig. 3—Ratio of the Heat Required at Any Point to the Heat Required at Top Dead-Center To Go from a Given Polytropic Curve to Another Very Close to It

Mr. Slonneger's method assumes that it requires the same amount of heat to go from a given polytropic to another at small crank-angles as it does at large crank-angles. Fig. 3 is presented for the purpose of showing the effect of the author's assumption on the amount of heat involved. It shows the ratio of the heat required at any point to the heat required at top center to go from a given polytropic to another very close to it. It will be seen that the heat required in going from one polytropic to another varies with the crank angle.

Another Analytic Method Claimed More Accurate

—Alfred T. Gregory

Wright Aeronautical Corp.

MR. SLONNEGER has made a very interesting and useful contribution to the study of the combustion in an internal-combustion engine. It is a far more practical method than the entropy diagram, as it is so very simple to use. However, I wish to point out that no two polytropics are parallel to each other in Mr. Slonneger's diagram and that his method of determining the end point of the combustion is therefore theoretically incorrect. For, if the combustion curve becomes parallel to a polytropic, it cannot itself be a polytropic of the same order. If a polytropic curve is found which coincides with the latter portion of the expansion line and which is a multiple of the first polytropic, the end point may be determined as the transition point from the combustion curve to the polytropic.

A little over two years ago while doing research work for a Doctor's thesis at the Technical College in Munich, Germany, I tried using the Stodola entropy diagram for combustion analysis. I soon found that the time required in using it and the inaccuracy involved by the number of unknowns which must be guessed at hardly justified its use. I therefore developed a method of analysis which proved to be very satisfactory and appears to me to be somewhat simpler and more accurate than the one Mr. Slonneger has proposed here.

It is a well-known fact that a polytropic expansion line, when plotted on logarithmic paper, is a straight line. The slope of the line gives the value of the exponent of the expansion of the polytropic equation:

$$pv^n = C \tag{1}$$

where p is the pressure; v, the volume; n, the polytropic exponent; and C, a constant dependent upon the individual conditions.

The value of n varies according to the type of expansion whether isothermal, adiabatic or some other type. The ease with which n can be measured on the logarithmic diagram makes that diagram a useful means of studying the combustion in an internal-combustion-engine cylinder. The rapidity of the combustion at any instant would be a function of the exponent n. Thus, when the instantaneous value of n is equal to the adiabatic exponent k, there will be no effective combustion going on in the cylinder.

Since the portion of the curve after the completion of the combustion will be a straight-line adiabatic, it is a simple matter to use the logarithmic diagram for determining the end point of the combustion. The expansion curve is taken from the indicator card and plotted on logarithmic paper. The straight-line adiabatic-portion of the curve is then pro-

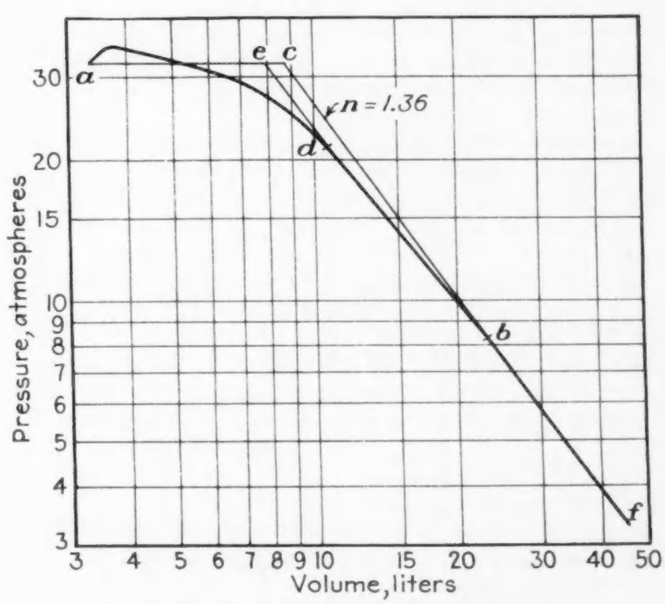


Fig. 4—An Expansion Line, *a d b f*, Taken from an Air-Injection Type Diesel Engine

duced so that the transition point from the curved to the straight-line portion can be more accurately determined. This transition point marks the end point of the combustion. It should be pointed out here that it is much easier to determine the transition point from a curve to a straight line than from one curve to another, so that the accuracy with this method would be considerably greater than with Mr. Slonneger's method.

The proportion of the fuel which has been effectively burned up to any point in the stroke may also be determined readily from the logarithmic diagram. A constant-pressure line must first be drawn on the logarithmic diagram from the ignition point and extended to intersect the adiabatic line, produced if necessary. Through the desired point on the expansion line then, a second adiabatic line must be drawn to intersect the constant-pressure line. The proportion of the fuel in the cylinder which has been effectively burned up to the desired point is given by the ratio of the volume increases along the constant-pressure line from the ignition point to each of the two adiabatics.

Fig. 4 shows an expansion line *a d b f*, taken from an air-injection type of Diesel engine. The ignition occurred at *a*, while the end point of the combustion was at *b*. The line *b f* is the adiabatic expansion line which has been extended to meet the constant pressure line drawn from *a* at *c*. The volume increase from *a* to *c* is 5.14 liters. Through any desired point *d*, a second adiabatic has been drawn parallel to *c f* and intersection *a c* in *e*. The volume increase from *a* to *e* is *x*. The proportion of the effective combustion at the point *d* is therefore given by the ratio x / 5.14.

The correctness of this method may be demonstrated as follows: For a constant-pressure expansion,

$$Q = c_p\,(T_2 - T_1) = c_p T_1 \left(\frac{V_2}{V_1} - 1\right) \qquad (2)$$

where Q is the heat developed per pound of combustible, c_p is the specific heat per pound of combustible, T_1 and T_2, V_1 and V_2 are the starting and final temperatures and cylinder volumes respectively. From Equation (2) it follows that

$$\frac{Q_c}{Q_e} = \frac{(V_c - V_a)}{(V_e - V_a)} \qquad (3)$$

that is, the heat supplied during a constant-pressure combustion is proportional to the volume increase when the starting conditions of temperature, pressure and volume are the same and the specific heats are equal. In Equation (3), Q_c is the heat added per pound during the expansion from *a* to *c*, Fig. 4, while Q_e represents the heat added per pound from *a* to *e*. V_c and V_e are the cylinder volumes corresponding to the points *c* and *e*, respectively, while V_a is the cylinder volume at the ignition point.

Air Utilization in the Hesselman Engine

IN a compression-ignition engine it is, for well known reasons, necessary to obtain ignition of the injected fuel almost immediately after injection. In order to obtain this rapid ignition, the fuel must reach its ignition temperature in a period of time corresponding to only a few degrees of crank-shaft travel. The fuel therefore must be injected into an atmosphere the temperature of which is well over the ignition temperature of the fuel, and this temperature is obtained by a high compression ratio. In a carburetor engine, on the other hand, all of the fuel is introduced into the cylinder during the suction stroke. In order to avoid spontaneous and uncontrolled ignition of the charge, it is necessary to keep the temperatures and consequently the compression ratio relatively low. In the carburetor engine the heating of the fuel is objectionable, and in the compression-ignition engine it is desirable. It is obvious that the heating of the fuel is a question of temperature, but it must not be overlooked that it is also a question of time. In a carburetor engine the fuel picks up heat from the walls of the combustion chamber during the 360 deg. of the suction and compression strokes, and is exposed to the heat of compression during the whole compression stroke.

Is it not possible to use a higher compression ratio in a spark-ignition engine if the time during which the fuel is exposed to heat is decreased? It is. This is, however, possible only in an injection-type engine, and it is one of the principles of the Hesselman engine. In this engine, fuel injection starts 50 to 60 deg. before top dead center, and the fuel is subjected to the heat of the combustion chamber during a relatively short time. Depending upon the fuel used, and the size and design of the engine, Hesselman engines can work with compression ratios of from 6:1 to 10:1.

The factor that limits the output of an engine of a given size is the air. It is possible to supply an almost unlimited amount of fuel to the engine, but the air supply is limited by the swept volume. It is therefore necessary to make the best possible use of the air. In a carburetor engine this is rather easy. Fuel is supplied to the cylinder during the suction stroke and has plenty of time in which to mix with the air and evaporate, and at the end of the compression stroke a homogeneous gas mixture is ready to be ignited.

In the compression-ignition engine the mixing of the fuel with the air is a comparatively difficult problem, and much of the development work on this type of engine has been concentrated on this point. It is necessary for the injected fuel to be evenly distributed throughout the air charge of the combustion chamber, and in order to obtain this distribution the combustion chamber has been formed so as to suit the shape of the spray, the air charge has been given a rotating or turbulent movement, or pre-combustion chambers have been employed. It is a well known fact, however, that the compression-ignition engine does not utilize the air very efficiently, and it is necessary to supply a considerable amount of excess air to this type of engine. The reason for this is obvious.

There are two factors controlling the mixing of the fuel with the air. One is space and the other is time. The first factor is usually well taken care of. The other factor, however, cannot be taken care of as easily in a compression-ignition engine because it is necessary for the proper operation of the engine that the fuel be burned as soon as possible after it comes in contact with the air in the combustion chamber. Only the first fuel injected meets fresh air.

—Excerpts fro ma 1934 Semi-Annual Meeting paper on "A High Power Spark Ignition Injection Engine" by Torbjorn Dillstrom, Hesselman Motor Corp.

Aircraft-Propeller Development and Testing Summarized

By Frank W. Caldwell

Research Engineer, Hamilton Standard Propeller Co.

Part 2

(Concluded from page 310, August issue)

THE aluminum-alloy propellers show an improvement in efficiency over the corresponding wooden propeller of about 5 to 7 per cent at tip speeds up to about 800 ft. per sec. This gain is partly the result of aspect ratio and reduced body interference, and partly the result of a thinner and more efficient airfoil. At higher tip speeds, the comparison becomes more favorable to the solid-metal propeller. At a tip speed of 1000 ft. per sec. for the metal propeller, it will show an advantage of about 15 per cent over the corresponding best wooden propeller, which will have a tip speed of about 920 ft. per sec.

The aluminum-alloy propellers are materially heavier than the wooden ones, though not quite as much heavier as might appear on the surface, since reduction ratios are usually selected to give a maximum tip-speed around 1000 ft. per sec.; whereas the wooden ones are usually geared to a tip speed of 800 ft. per sec., so that a larger propeller results.

The wooden propellers certainly have an advantage as to noise. The controllable-pitch propeller gives some improvement as to noise by permitting cruising at a somewhat lower number of revolutions per minute, as discussed under the subject of controllable-pitch propellers.

If we were to rate the five materials according to some of the desirable properties of propeller material, the relative merits would be in about the order given in the following tabulation of propeller types, the steel rating being based on the assumption of hollow steel-blades.

Specific Gravity	Specific Ultimate Tensile Strength	Specific Fatigue Strength	Internal Damping	Resistance to Abrasion	Resistance to Weathering
Wood	Wood	Wood	Micarta	Steel	Micarta
Micarta	Magnesium Alloy	Magnesium Alloy	Wood	Aluminum Alloy	Aluminum Alloy
Magnesium Alloy	Aluminum Alloy	Micarta	Magnesium Alloy	Magnesium Alloy	Steel
Aluminum Alloy	Steel	Steel	Steel	Micarta	Magnesium Alloy
Steel	Micarta	Aluminum Alloy	Aluminum Alloy	Wood	Wood

At the present state of development, I should rate the advantages of the actual propellers about as in the following tabulation, remembering that this rating applies to our present knowledge of the art.

Noise	Weight	Dependability	Efficiency	Repairability	Wearing Properties
Micarta	Wood	Aluminum Alloy	Aluminum Alloy	Aluminum Alloy	Steel
Wood	Micarta	Micarta	Magnesium Alloy	Magnesium Alloy	Aluminum Alloy
Magnesium Alloy	Magnesium Alloy	Wood	Micarta	Steel	Micarta
Aluminum Alloy	Aluminum Alloy	Magnesium Alloy	Steel	Wood	Magnesium Alloy
Steel	Steel	Steel	Wood	Micarta	Wood

Some Types of Controllable-Pitch Propellers

We have built controllable-pitch propellers with blades of wood, micarta, hollow steel and aluminum alloy. Examples of each type which have been developed under the direct supervision of the author are illustrated in the following brief description.

Fig. 36 shows the general proportions of the type of controllable-pitch propeller with wooden blades and manual control as worked out during 1920. As shown, the blade ends were anchored in steel ferrules, the ferrules being pressed on with a pressure of about 30 or 40 tons. At the time the ferrules were pressed on, the blade ends were coated with shellac so that some additional adhesion was obtained. After the ferrules were pressed on the blade a small cylindrical wedge, shown at 3, was pressed in the ends of the blades, forcing the wood out against the reverse taper. This wedge was held in place by the small dowel, 4. In addition, two larger dowels at 2 and 5 were pressed through the shell of the ferrule and the wooden blade-end, providing additional anchorage. As shown, the pitch change was accomplished by a yoke carrying a ball bearing at the hub end, the outer race of this bearing being actuated by two screws, 61 and 62, which were driven through spiral gears by a cross-shaft, 52. This, in turn, was driven by a sprocket, 85, connected to the cock-pit control through suitable chains. The sprocket support, 75, contains an automatic locking-device to prevent creeping.

Fig. 37 shows a later type of controllable-pitch propeller, still using wooden blades, in which a ball-bearing mounting was placed inside of the blade ferrule, permitting the use of commercial sizes of bearings. It will be noted that, in this case, the propeller was provided with counterweights which

[This paper was presented at the Semi-Annual Meeting of the Society, Saranac Inn, N. Y., June, 1934.]

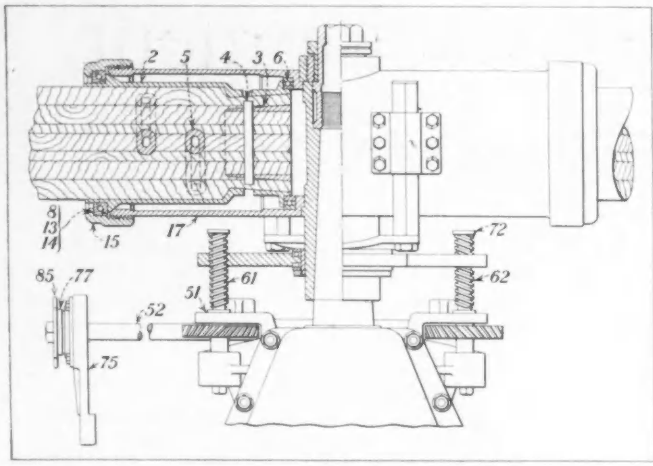


Fig. 36—A Later Wooden Type, the 1920 Hart Reversible Propeller

The mechanism is shown as applied to the 150-hp. Hispano engine

2	Ferrule	51	Intermediate Connecting Bracket
3	Expanding Wedge	52	Intermediate Connecting Shaft
4	Wedge-Retaining Pin	61	Control Screw, R. H.
5	Dowel Pin	62	Control Screw, L. H.
6	Blade-Base Bearing	72	Screw Cap
8, 13, 14	Blade Retainer Bearing	75	Control Shaft Outer-Support
15	Blade Retaining Collar	77	Locking Clutch Cage
17	Hub Barrel	85	Front Sprocket

were calculated to balance the centrifugal twisting-moment. This type was also actuated by a manual control.

Fig. 38 shows a controllable-pitch propeller with hollow steel-blades in which the blade ends are mounted externally around an internal arm carrying all of the bearings for taking up bending loads and centrifugal force. Several of these propellers were built and subjected to whirling tests, and engine tests, and used in flight tests. While this propeller was fairly satisfactory insofar as the tests revealed, the hollow steel-blade design was not considered satisfactory for production; so, only a few propellers were built.

Fig. 39 shows a controllable-pitch propeller with micarta blades. In this propeller, the bearings were all placed inside of the steel ferrule and the blade end was screwed into the ferrule. On account of the very light weight of the micarta blades, the bearing construction of this propeller was quite satisfactory. This propeller was also fitted with counter-weights for balancing the centrifugal twisting-moment.

In all of the foregoing propellers a manually actuated type of control was used for changing the pitch, and a ball bearing was used between the pitch actuating-yoke and the control parts. This ball bearing was found to wear quite rapidly in all cases, due to the unsatisfactory mounting. A very troublesome fitting job was required for each individual airplane to line up the control parts with the aircraft structure. This proved to be a rather unsatisfactory maintenance and installation job.

Figs. 40 to 47 show views of a two-way and three-way controllable-pitch propeller of the present Hamilton Standard type. These propellers have aluminum-alloy blades with a solid outer-portion and a hollow construction near the hub. The bending loads are taken up on plain bearings between the spider part attached to the crankshaft and the bore of the blade ends. The blade ends are provided with lead bronze-lined steel-bushings and an Alemite greasing system for keeping the bearings lubricated.

The thrust bearings taking the centrifugal force are placed around the outside of the blade end, as this makes possible a large roller-bearing area without increasing the size of the blade end excessively. The inner race of this thrust bearing has a large radius mating with a very large fillet on the end of the blade. This construction was found to last about seven times as long under vibration tests as the original de-

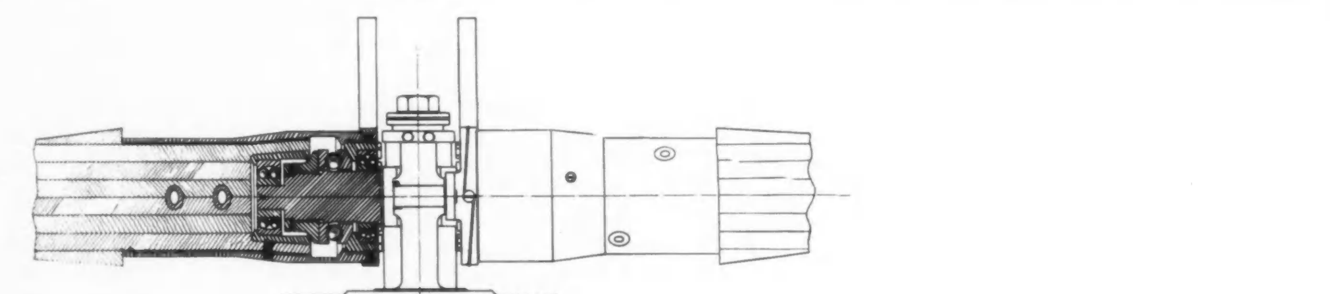


Fig. 37—Controllable-Pitch Propeller with Wooden Blades Built During 1920 for the Wright 300-Hp. Engine

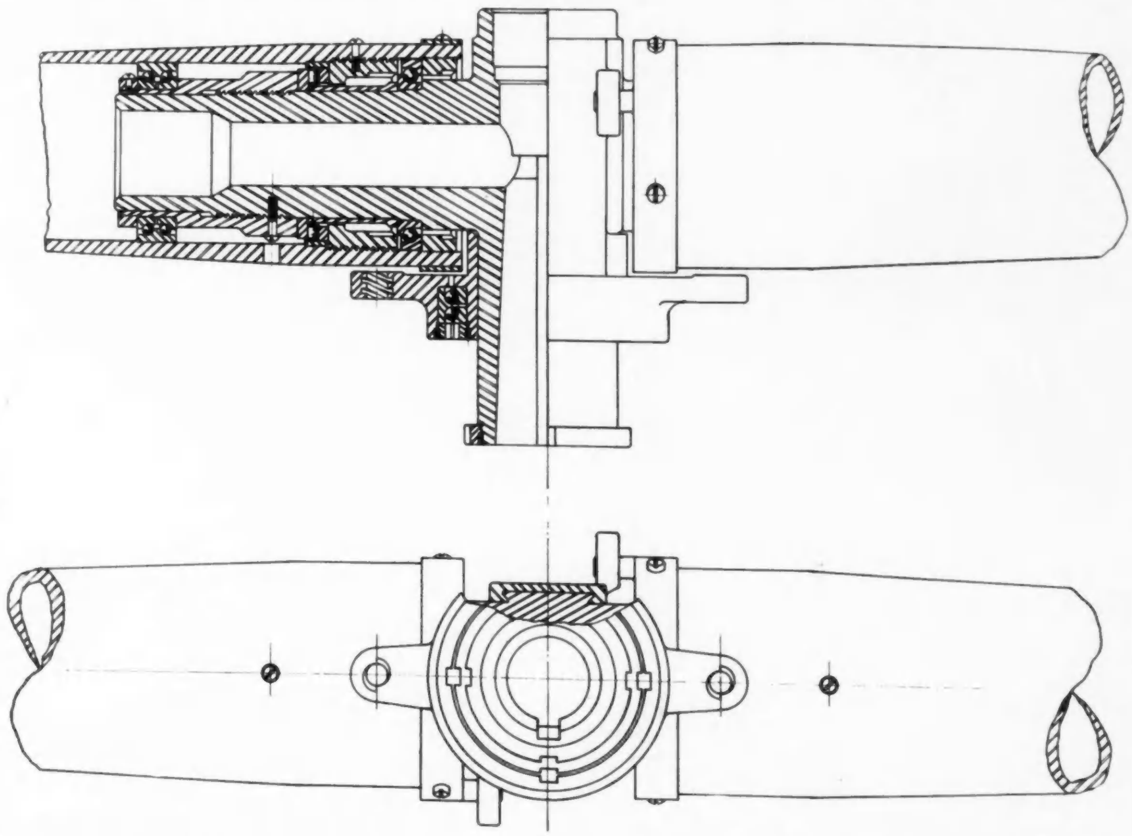


Fig. 38—Controllable-Pitch Propeller with Hollow Steel-Blades

sign, in which the bearings were seated against a flat shoulder with a moderate-sized fillet provided at the blade juncture. The bearing races are made of a special alloy-steel which may be heated to 1000 deg. fahr. without reducing the hardness of the rings. This is necessary because the blades have to be heat-treated after the rings are in place.

To assemble the races on the blades, the blade ends are first made up in a solid cylindrical form with a small enough diameter to permit sliding the races over the ends of the blades. The blades are then heated up to forging temperature, the races slid on to the blades, and the blade ends are upset in a forging machine with the races in place. In the machining operation, the races are pushed back out of the way and the blade end machined to size.

The propeller control is of the hydraulic type, using the oil from the lubrication system of the engine. To simplify the design as much as possible and to eliminate the necessity for small oil-leads which might cause trouble in service, we use a single-acting piston to change pitch by means of oil pressure in one direction only and rely on centrifugal force to shift the pitch in the opposite direction. To further simplify the operating problem, we provide limit stops at the ends of the stroke and use the propeller as a two-pitch type. The two positive settings have proved to be a very great advantage during the period while the operating and maintenance personnel have become familiar with the use of controllable-pitch propellers. These propellers have been built in sizes ranging from a diameter of 8 ft., suitable for the 100-hp. engines, up to a diameter of 13 ft., suitable for the geared engines of 800 hp.

To simplify the transition from the ground adjustable type, which we have been building in large quantities for a number of years, we have set up a series of standard controllable-pitch blade-ends designed to replace the corresponding standard ends of the ground adjustable type. The corresponding standards are as follows:

Ground Adjustable Blade-Ends	Corresponding Controllable-Pitch Blade-Ends
0	*A*
½	*B*
1	*C*
1½	*D*
2	*E*
3	*F*

We have made overload whirling-tests, engine tests, and destructive vibration-tests on the controllable-pitch blade-ends and compared them with the corresponding standard blade-ends so as to assure in each case that the controllable-pitch blade-end is stronger than the standard blade-end which it replaces. By following this procedure, we have been able to go into the production quite rapidly on the controllable-pitch propellers without fear of failures and, up to March 1, 1934, we had a total of over 1000 controllable-pitch propellers in use or released for construction. The largest quantity in actual-service use is the two-blade type for the direct-drive Wasp engine. About 200 of these are in use, and those on the United Airlines alone have accumulated over 75,000 hr. of flying.

Propeller Testing for Determination of Strength

By far the most difficult problem in connection with the propeller design is the determination of the actual degree of safety of the propeller during a continued period of use. The strength problem in connection with the wooden propeller is rather simple, as we do not have fatigue to deal with

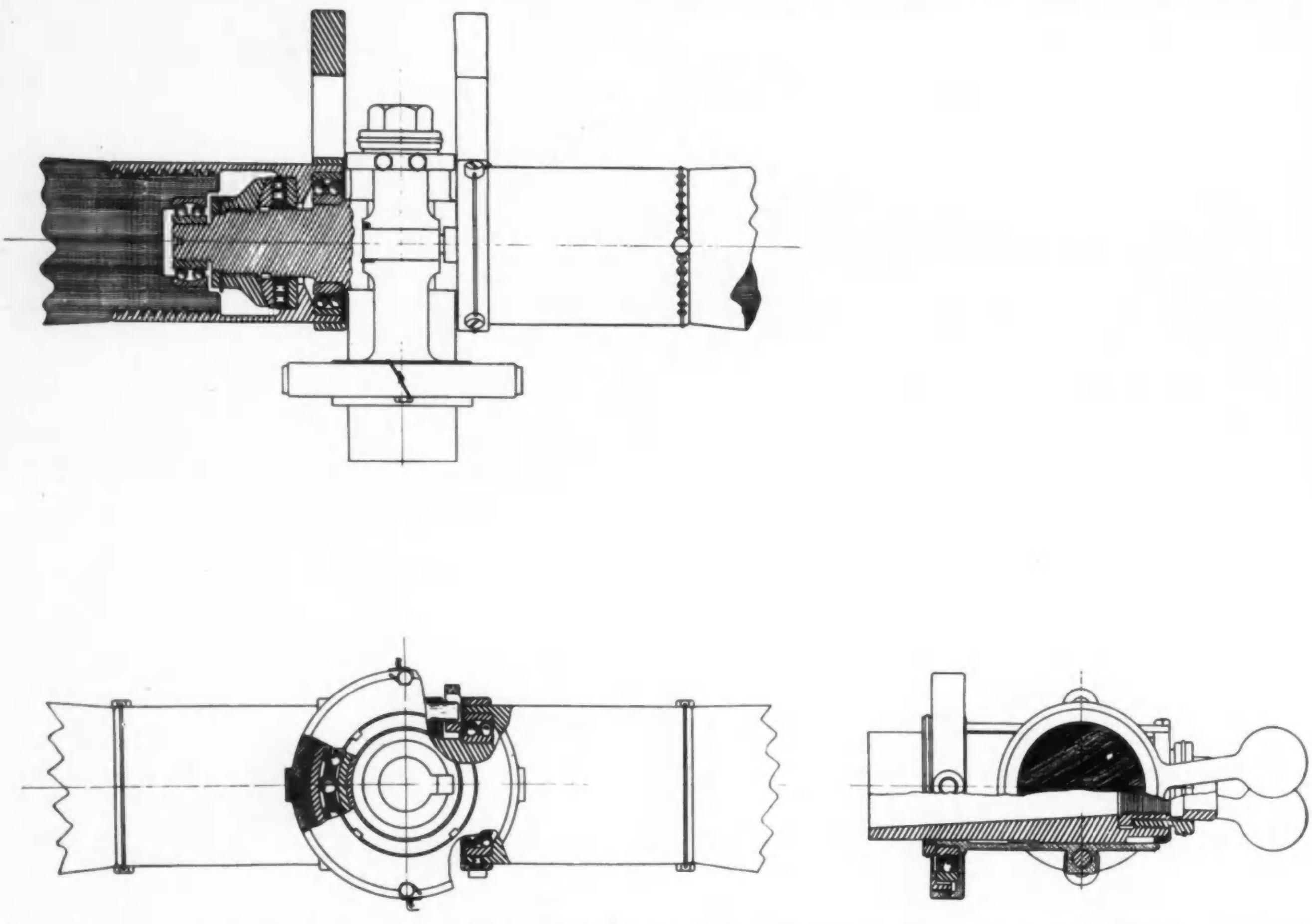


Fig. 39—Controllable-Pitch Propeller with Micarta Blades

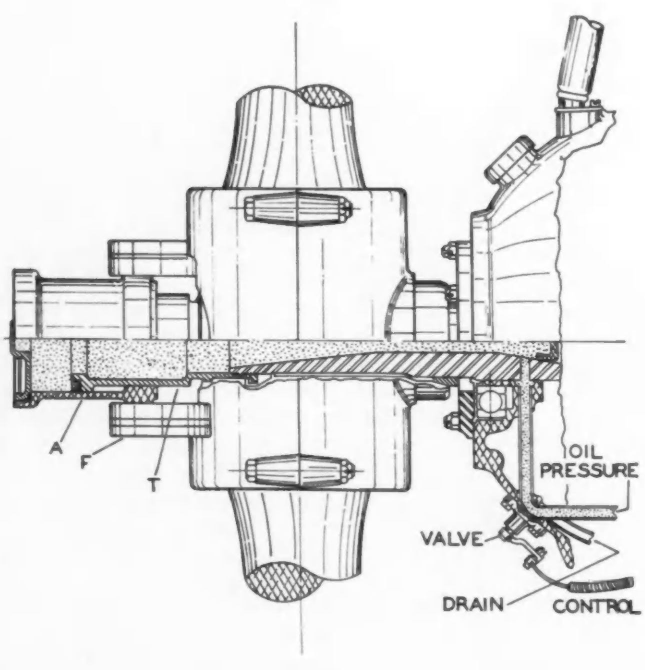


Fig. 40—Present Propeller, Showing Oil Supply

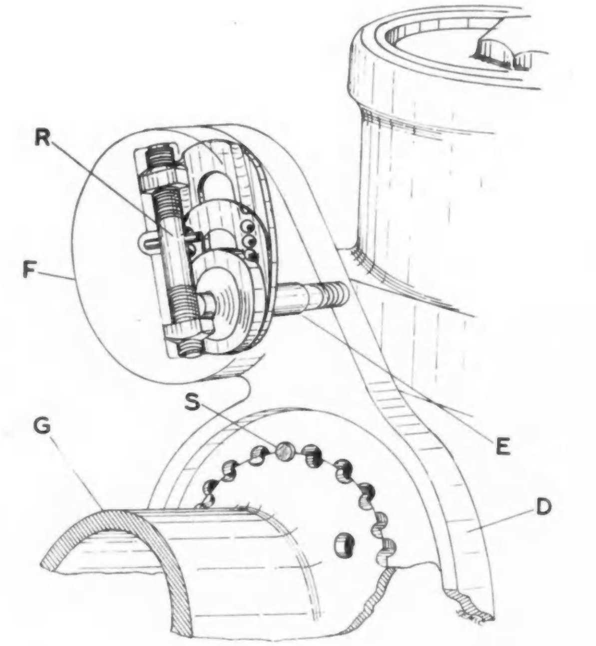


Fig. 41—Present Propeller, Showing Counterweight Bearings

During 1918, we adopted the practice of making a type test on each new design of wooden propeller and set up a standard test-requirement of a run of 10 hr. at a power input 50 per cent in excess of the rated horsepower of the engine on which it was to be used. This test was run on a whirling-stand powered by an electric motor. Three whirling-stands for this purpose have been used successively in the United States. The first, located in the plant of the Westinghouse Co., consisted of a jackshaft mounted in heavy pillow-blocks and driven by an electric motor through a belt. This stand had a power-input capacity of about 800 hp. and a maximum speed of 2500 r.p.m. The second whirling-stand set up, at the Army Laboratory at McCook Field, had a capacity of 1400 hp. at 2500 r.p.m. The power was furnished by four Sprague dynamometers connected in tandem to a jackshaft and equipped with scales for weighing thrust and torque.

The present testing laboratory at Wright Field is equipped with three stands, one rated at 2500 hp. with a maximum speed of 5000 r.p.m., a second rated at 4000 hp. with a maximum speed of 1800 r.p.m., and a third rated at 6000 hp. with a maximum speed of 900 r.p.m. All three stands are set at a height sufficient to swing propellers with a diameter of 50 ft. All three stands are also equipped with accurate means of reading thrust and power input.

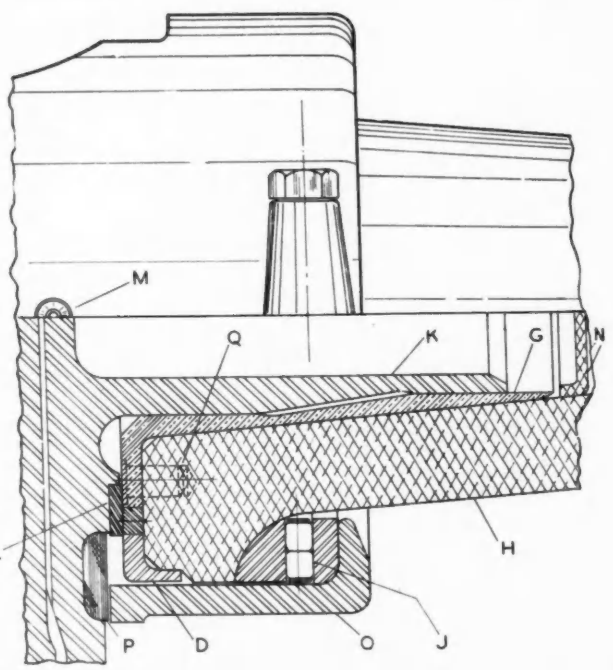


Fig. 42—Present Propeller, Showing Blade Mounting

The whirling-test requirements for metal propellers have been gradually increased until a test is now usually made at a power input equal to three times the rated horsepower of the engine for which the propeller is designed. While this test is very valuable it is not sufficient to prove the safety of the propeller, as the power impulses and other vibrations of the engine introduce very high vibration stresses independently of the centrifugal and air-pressure loads. For this reason, it is now required that the propeller shall be run on the engine for which it is designed for a period of 50 hr. at full throttle.

In building large numbers of propellers to operate over long periods of time without failures, we have not found the 50-hr. test quite sufficient to guarantee entire freedom from failures. We have also found it quite expensive and slow to try to work out a series of improvements through engine tests, particularly in view of the large number of engines destroyed in the process.

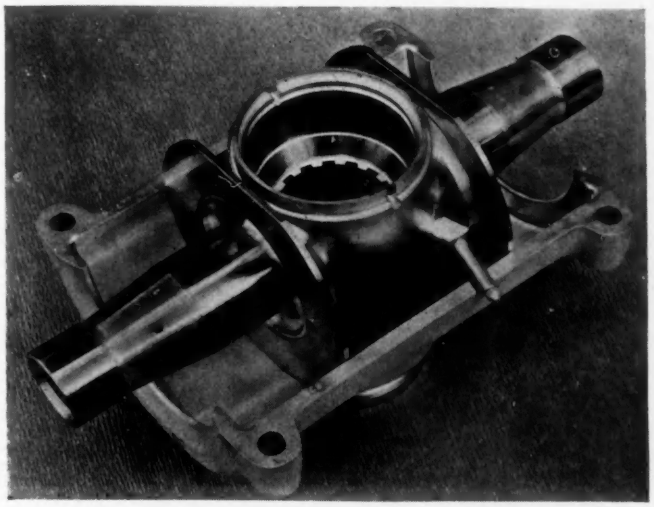


Fig. 43— Hub Spider and Half of the Retaining Barrel

For the purpose of making accelerated tests of the effect of forced vibration under severe overload, we have designed and built a machine in which the propellers are fastened at the hub to a reciprocating arm and shaken violently at right angles to their length. See Figs. 48 and 49. The machine operates at 1750 cycles per min. and the stroke of the reciprocating motion is about ⅜ in. This movement does not give sufficient loading to break the blades in a reasonable length of time; so, weights are attached to the blades at various points along the radius. These weights are adjusted roughly to suit the original weight-gradient of the blade. By variation of the amount of weight applied, the time required before failure can be controlled. In the actual use of the

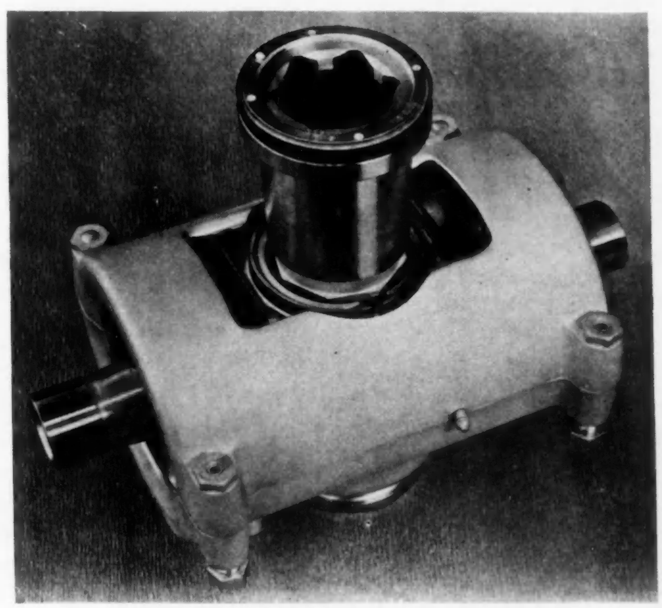


Fig. 44—Hub and Piston Assembly

Fig. 45—Complete Two-Blade Hub-Assembly

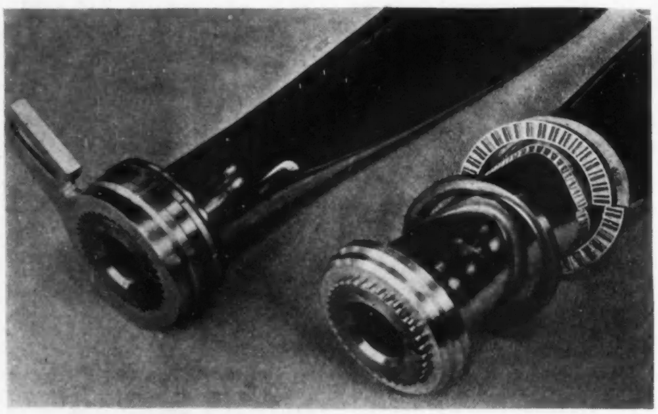


Fig. 46—Blade Assembly with Bushings and Thrust Bearings

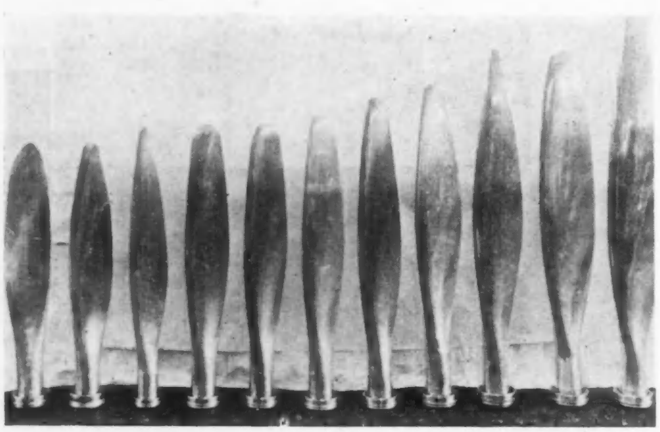


Fig. 47—Controllable-Pitch Blades from 200-Hp. to 800-Hp. Types

machine a standard is necessary. For this purpose, we select a propeller which has a record of long service without failure on a given type of engine. The load on this propeller is then adjusted until failure occurs after about 10 hr. The 10-hr. period corresponds to about one million cycles of stress, and is the minimum time for any comparison of fatigue effects. The loading and time of test for this particular propeller is now adopted as standard for the engine under consideration, and new types are required to equal or exceed the standard in resistance to the vibration test.

After working out the standard test for a number of engines and propellers of widely varying size, we have adopted a sort of formula for the loading which should produce a

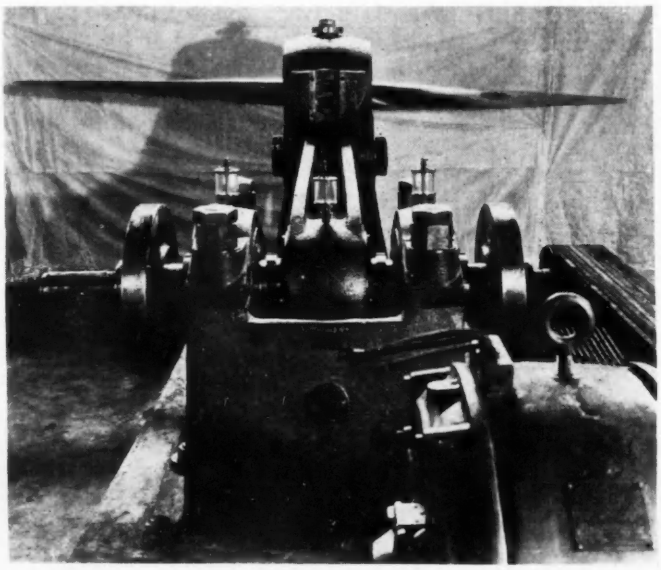


Fig. 48—Vibrating Machine and Driving Motor for Destructive Tests of Propellers

Fig. 49—Side View of Vibrating Machine Shown in Fig. 48

calculated bending-moment equal to the product of the square of the cylinder bore of the engine, multiplied by the distance from the shaft axis to the center of gravity of the blade, multiplied by a constant. This constant will depend on the method of mounting the machine, as previously explained.

The selection of the square of the cylinder bore is not entirely logical as a measure of the severity of the engine, because the maximum cylinder-pressure and the stiffness of the crankshaft and case also come into the problem. How-

ever, most engines are improved until they reach about the same mean effective pressure, and the mechanical engineering brings the other factors into some relation to cylinder size so that the bore is considered the best single practical criterion.

Our experience has been that the single-row radial-engines

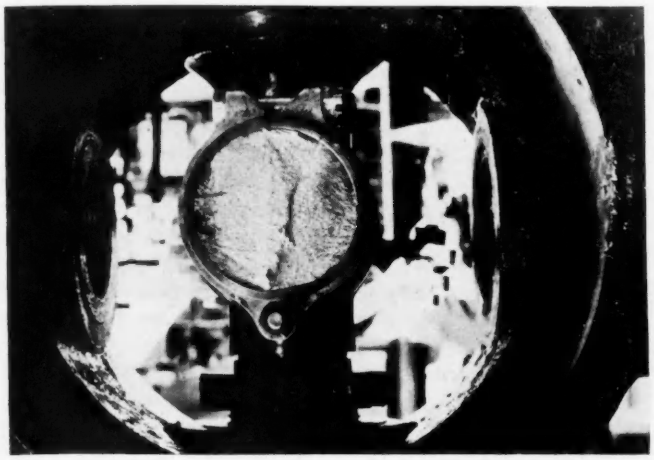


Fig. 50—Example of Vibration Failure of an Aluminum-Alloy Blade

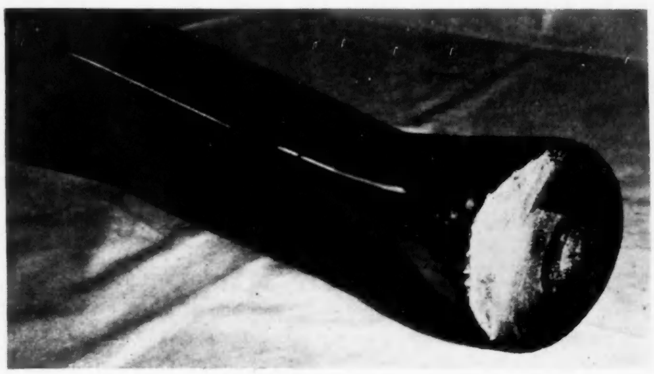


Fig. 51—Example of Vibration Failure at Blade Root on the Testing Machine

stress the propellers much more severely than do the 12-cylinder V-type at the same rating. This is partly due to the larger size of individual cylinder of the single-row types and partly to the more irregular torque-curve. It is thought to be mainly due, however, to the greater distance between the propeller and the cylinders of the V-type engine, resulting in a softening of the blow from the cylinder explosions both torsionally and laterally.

A number of blades and hubs have been broken on this machine and, in general, the results have been in fair agreement with the failures which occur on engine test and in service. The results would, no doubt, be somewhat altered by the presence of centrifugal force, but the machine would be much more complicated and difficult to operate.

Some examples of fractures produced on the vibrating machine are shown in Figs. 50 to 56 inclusive. These tests show failures of the conventional aluminum-alloy propellers, of a hollow steel-construction and of a magnesium-alloy blade. A number of tests of the new controllable-pitch propellers have been carried out on the vibrating machine with the same loading as for the corresponding propellers of the old type. So far, no failures have been produced, as these pro-

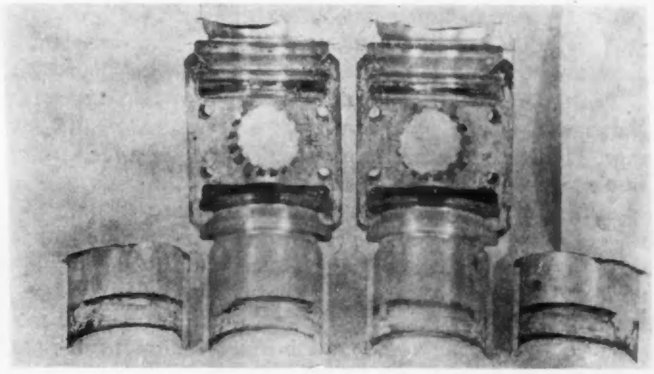


Fig. 52—Failure of a Second Type of Hub on the Testing Machine

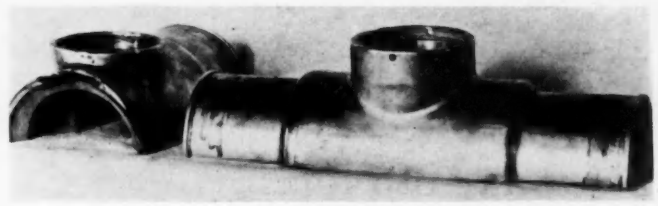


Fig. 53—Failure of a Third Type of Hub on the Testing Machine

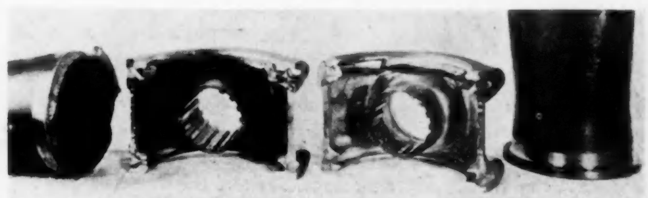


Fig. 54—Failure of Hollow Steel-Hub and Blade on the Testing Machine

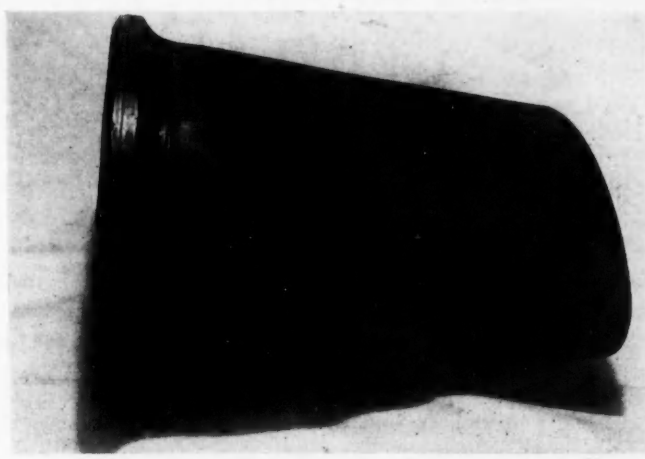


Fig. 55—Failure of Another Type of Hollow Steel-Blade on the Testing Machine. Date, Aug. 17, 1932. Test No. 39. Time, 283 min. Design of Blade with 5-In. Insert, 40923-6. Design of Hub, Special

Fig. 56—Failure of Second Type of Magnesium-Alloy Propeller-Blade on the Vibrating Machine

pellers appear to have a much higher resistance. The tests will be continued to destruction, however, as a means of exploring the possibility of lighter weight.

With the present machine, the results are influenced by the rigidity of the mount and by the clearance in the bearings. These two factors have to be kept constant for uniform results. This device is by far the most satisfactory method of testing the strength of the hub attachment and inner portion of the blades, and we have been able to eliminate the major failures since adopting this criterion. The machine does not test the outer parts of the blades, and we are relying largely on our resonant-vibration studies and stress measurements to eliminate the tip failures. Conditions of maintenance will always have a great bearing on the tip failures, however, and prompt removal and rounding out of nicks in service are essential.

Discussion

Suitability of Propeller-Blade Materials Set Forth

—G. T. Lampton
Lycoming Mfg. Co.

MR. CALDWELL'S excellent paper gives a very fine and comprehensive picture of the present state of the art of propeller design, but I am forced to disagree with some of his comparisons of propeller-blade materials. I agree that the ratio of the tensile properties to density is an excellent rough criterion of the suitability of the materials. The figures for the minimum tensile-strengths specified for steel and dural blades are:

Property	Steel, 6130	Dural, 17-St
Ultimate tensile-strength	135,000	55,000
Yield point	115,000	30,000
Endurance limit	55,000	14,000
Density, lb. per cu. in.	.283	.101
Specific ultimate-strength	477,000	545,000
Specific yield-point	406,000	297,000
Specific endurance-limit	195,000	139,000

The specific endurance-limit indicates the order of merit with regard to blade weight, though it cannot be realized in the direct-drive diameters due to practical limitations on the tip wall-thickness. The specific yield-point indicates the ability of the blades to withstand high-revolutions-per-minute dives, which has become an important part of the acceptance tests of new military aircraft. The advantage is obviously with the steel blades on both counts.

The matter of noise, weight, dependability and efficiency are so closely interrelated that they can hardly be discussed or classified separately. Steel blades are generally stiffer in bending and torsion than are their aluminum-alloy equivalents. Therefore, they vibrate with smaller amplitudes at frequencies of the same order, and hence should be better than aluminum-alloy blades so far as noises of mechanical origin are concerned. Whether the noises arising from tip vortices and the like will be worse or not will depend on the camber ratio in the neighborhood of the tip, the angle of attack and the tip velocity, and these factors are determined by the particular design and application. Personally, I have never observed any difference in noisiness between the two types, though a difference might be detectable by acoustic measurement.

In the hollow steel-blade, the designer has the option of employing high camber-ratios and thin walls, or the reverse. In the former case the blades will be slightly less efficient but lighter; or, on the other hand, the maximum efficiency can be achieved at the expense of some added weight. In actual practice, the majority of the hollow steel-blades in service today are about 2 lb. heavier than their dural equivalents and, being external duplicates of standard dural designs, have identical efficiencies. The fact that the possible weight saving in the large-diameter hollow steel-blades has not been realized shows that these designs are very conservative and have larger margins of safety than do those of the dural type. With the exception of one design which we voluntarily retired from service because of bad vibration characteristics, this statement is supported by the service history of nearly 3000 blades. I therefore presume that, when Mr. Caldwell rates the steel blade last in dependability, he refers to the experimental blades of the early 1920's and not to the atomic-hydrogen-welded blades of current production.

It is true that the high tooling-charges of steel blades do not allow as large a number of designs as are available in aluminum alloy and that this may result in a blade selection which is a less-exact fit for some airplane-engine combinations, with a slight loss in efficiency. This is a negligible factor, however, when compared with the ability of a continuous-pitch controllable-assembly to operate at the maximum allowable engine limits regardless of flight regime and altitude.

Merits of Hollow Steel-Propellers Presented

—Hamilton Foley
Pittsburgh Screw & Bolt Corp.

MR. CALDWELL'S explanation of his controllable-pitch propeller makes a most important chapter on United States aviation-engineering history. His explanation of what has been and is to be done to have aluminum-alloy propeller-blades safe in service will be studied by experts everywhere. What he has not said in his paper—about one topic—is of importance and calls for a word from him.

Mr. Caldwell's statements about hollow steel-propellers are surprising in view of all the facts set forth in my paper on the Manufacture and Testing of Hollow Steel-Propellers, read at the International Automotive Engineering Congress Meeting of the Society in 1933. His unqualified statements that "we still need to learn more before we can adapt them to aircraft in a responsible way as far as safety is concerned" and that we are "further from the expected saving in weight" are particularly surprising when it is noted that hollow steel-blades manufactured in the United States have had full U. S. Government approval for design, for efficiency and for safety, over a period of years.

Dicks-type hollow steel-blades were designed for and have been used with safety and efficiency on U. S. Army and Navy aircraft for nearly four years. Mr. Caldwell states that "the first aluminum-alloy propeller blade was designed from the information which we had obtained from our solid-steel experience." He notes how that first solid aluminum-alloy blade has been duplicated all over the world and is still in use. The first Dicks-type hollow steel-blade was designed from the information which we had obtained from experience with the

solid aluminum-alloy blade. Before the first Dicks hollow steel-blade was manufactured a careful stress analysis of the design was made by Commander Havill, the propeller officer of the U. S. Navy, who a few weeks later, received an international award for his propeller studies. After the propeller officers of the U. S. Army Air Corps had approved the design for the first Dicks hollow steel-blade, it was manufactured and tested. That first blade-design is now standard equipment with the U. S. Army Air Corps. All subsequent Dicks hollow steel-blade designs have received the official approval of the propeller engineers of the Air Forces of the U. S. Government before being put into steel. The U. S. Department of Commerce officially approved Dicks-type hollow steel-blades for passenger and commercial aircraft, over three years ago.

All of this Government approval was preceded necessarily by all the detailed and careful official Government tests given to all metal-propeller blades. Dicks hollow steel-blades, however, are subjected to and pass an additional and more drastic test than is imposed on any aluminum-alloy blades. No Dicks hollow steel-blade is offered to or accepted by the U. S. Army or Navy until it has been examined by the manufacturer and by the U. S. Government inspectors under the "magnaflux" test, which determines whether there be any flaw in the blade invisible to the eye and in many cases to the X-ray that might be the cause of weakness or the point of origin of a blade failure in flight. Each and every Dicks-type hollow steel-blade made for commercial service is similarly examined and approved by the manufacturer and by the U. S. Department of Commerce before that blade is released for commercial or private flight-service. Dicks hollow steel-blades may be magnafluxed at frequent intervals within a few minutes, to determine whether any internal flaw be developing. Aluminum-alloy blades cannot be examined in this way by any test as practical and as dependable.

Hollow steel-blades of the Dicks type have recently passed all the new and more drastic vibration, whirl, engine and magnaflux tests adopted by the U. S. Army Air Corps within the last three or four months. They have been designed and manufactured for use with hubs of the adjustable-pitch and for hubs of the controllable-pitch type. They have Government approval for use with both types of hub.

Mr. Caldwell proves that this is the day for controllable-pitch propellers. The development of controllable-pitch propellers was responsible to a large extent for the continued encouragement given by the U. S. Army Air Corps to the improvement of, and for the production scale manufacture of, Dicks hollow steel-blades. No propeller engineer needs to be told why.

Mr. Caldwell writes without approval of the weight of hollow steel-blades. Controllable-pitch propellers approved for use by the U. S. Army Air Corps and by the U. S. Navy, weigh less with Dicks hollow steel-blades than with aluminum-alloy blades.

Mr. Caldwell states that the largest quantity of controllable-pitch propellers with aluminum-alloy blades in actual service is the two-bladed propeller for the Wasp direct-drive engine. Two-bladed controllable-pitch propellers in size suitable for U. S. Air Corps or U. S. Navy use, with supercharged "Wasp" engines, complete with controls, with Dicks hollow steel-blades, weigh 121 lb. The controllable-pitch propeller with aluminum-alloy blades for the same service weighs 159 lb. The three-bladed controllable-pitch propeller—which is 11 ft. in diameter—now on special test by the U. S. Army Air Corps for proposed use with either the "Hornet" or the "Cyclone" engine, equipped with Dicks hollow steel-blades, weighs 285½ lb. A controllable-pitch propeller of the same diameter now being tested by the U. S. Army Air Corps for proposed use with the same engines, equipped with aluminum-alloy blades, weighs 330 lb.

There is no practical hope that the weight of the aluminum-alloy blades used in these controllable-pitch propellers may be reduced. There is reason to believe that the weight of the Dicks hollow steel-blades—as now manufactured—may be reduced appreciably, and the U. S. Air Corps propeller engineers, of their own initiative, are now studying modifications of design that will reduce the weight of these blades.

Mr. Caldwell, writing of aluminum-alloy blades, states that "we are relying largely on our resonant-vibration studies and stress measurements to eliminate tip failures." He mentions at least three times in his paper the dangers from tip failures.

Before the propeller engineers of the U. S. Government had formulated definite conclusions about resonant vibration in metal blades, two different blade-designs of Dicks hollow steel-blades had been approved by the U. S. Army and Navy and the Department of Commerce. Since resonant-vibration studies have become better understood, four new blade-designs for Dicks hollow steel-blades have been designed by the propeller engineers of the U. S. Army Air Corps, with special regard to vibration studies. These new designs have been put into metal and have passed Government vibration, whirl, magnaflux and engine tests. Flight service of the first two blade-designs has been restricted so as to keep them clear of the vibration periods now known to be dangerous for those particular designs.

Mr. Caldwell, writing of aluminum-alloy blades, states that there have been "a number of tip failures" and that these tip failures "are found to be started at a nick." He makes several references to the dangers from nicks in blades. He also emphasizes the fact that "conditions of maintenance will always have a great bearing on tip failures, and that prompt rounding-out of nicks in service is essential."

Hollow steel-blades of the Dicks type have practically entirely eliminated the "conditions of maintenance" Mr. Caldwell specifies and which are as well known in connection with aluminum-alloy blades. The Dicks-type hollow steel-blades do not "nick" in service. This fact is demonstrated by the condition of the hundreds of Dicks hollow steel-blades in Army and Navy service, many of which now have an average of 500 hr. or more in flight.

Dicks hollow steel-blades in commercial service on passenger planes operating overland have over 1000 hr. of flying service without what would be called a "nick" in the blades in the sense in which Mr. Caldwell uses that word.

The commercial planes with which Dicks hollow steel-blades have made this "maintenance" record operate on fields on which gravel caused the objectionable "conditions of maintenance" to aluminum-alloy blades at least every 300 hr.

Dicks hollow steel-blades on Army, Navy, and commercial seaplanes operating over salt water, have demonstrated that there is none of the pitting or corrosion that materially affects aluminum-alloy blades in any time between 5 and 50 hr. of flight service. When it is remembered that an aluminum-alloy blade may receive a "nick" at any moment, no words are necessary to prove how advantageous from the safety and the economy consideration are hollow steel-blades that do not nick in service, provided that the steel blades have full Government approval.

When Mr. Caldwell uses the words "aluminum-alloy,"

"magnesium-alloy," "micarta" or "wood" propellers, all aviation men know to what type he refers. There is only one type of each of these materials. When he speaks merely of hollow steel-propellers, what he means is not clear. Many kinds of hollow steel-propellers have been tested and failed. Only one type has ever received U. S. Government approval for flight service on commercial, passenger and mail planes. This one exception is the Dicks-type hollow steel-blade. Because this type has received such complete U. S. Government approval, because it has special usefulness for controllable-pitch propellers and has proved in Government service so free from the maintenance conditions that have proved both dangerous and expensive, Mr. Caldwell's unqualified statements condemning hollow steel-blades in general do not accord with the facts of record of the Dicks-type hollow steel-blades. His unqualified statements condemning all hollow steel-blades ignore the fact that American aviation-engineering talent has made hollow steel-blades for over four years with the full official approval and use by the Air Forces of the Government of the United States. As Mr. Caldwell knows how exactingly a new propeller is tested before it is used by the Air Forces of his Government, the unqualified statements in his paper, condemning all hollow steel-blades, are as surprising as they are inaccurate.

Relative Advantage-Ratings Criticised

—D. A. Dickey

MR. CALDWELL'S paper is commendable for its summary of information of general interest, but those portions dealing with his studies of distribution of stresses in vibrating propellers is of particular and timely interest to the propeller engineer who today is struggling with this agonizing complex—vibraphobia, as it were—arising from the perfectly natural phenomenon of vibration. It is believed, however, that there are certain portions of the paper that may lead to misunderstanding and result in the deduction of wrong conclusions by those not in possession of certain facts. This opportunity is taken to direct friendly criticisms at those portions of the paper.

In his remarks about steel blades, it is not clear whether Mr. Caldwell has in mind some certain types with which he is particularly familiar or whether he is referring generally to all types of steel propellers as one would be led to infer from his opening remark; "It is the purpose of this paper to outline in the briefest possible way some of the experiments with aircraft propellers carried out *in the United States* during the last few years." To say that results have been disappointing without telling what the expectations were is really telling very little. Expectations may have been too high and, in spite of disappointments, a commendable measure of practical success may still be experienced. If the author is referring to hollow steel-propellers generally, he is decidedly unfair in saying that: "We still need to learn more before we can design and adapt them to aircraft in a responsible way as far as safety is concerned." Quite a respectable number of hollow steel-propellers, properly applied, are giving unusually satisfactory and safe service today. The most serious problems that beset the hollow steel-propeller insofar as safety is concerned are the same as those which beset the solid aluminum-alloy propeller; namely, resonant vibration of the propeller and torsional resonance of the propeller shaft.

In this same connection there is room for legitimate argument as to the relative advantage-ratings he has given steel propellers, particularly in dependability and efficiency. Steel propellers are rated below micarta and wood propellers in dependability. Both of these latter are admittedly obsolete for present-day engines of high power; also, steel propellers are rated below magnesium-alloy propellers which, as a class, are too experimental today to even merit any rating at all. And insofar as efficiency is concerned, there are tests which show both hollow-steel and solid-steel propellers equalling if not actually surpassing in efficiency a comparable aluminum-alloy propeller of conventional design. There seems to be nothing peculiar to steel to bar it from equalling aluminum alloy in efficiency.

One more consideration that cannot be passed without a friendly challenge is that of the relative noise-ratings. The really important factors responsible for the propeller noise that concerns us today are, principally, the peripheral velocity of the tips; to a noticeable extent the thickness of the blades; and to some uncertain extent the pitch of the sound. The relative ratings as listed ignore consideration of these factors and are seriously misleading to the uninitiated. In fact, I am sure that Mr. Caldwell would be one of the first to admit that his table of relative ratings of various types of propellers is not a table of truly engineering information.

Phases in Railcar Development

During the period from 1921 through 1929, the railcar was developed from a single unit to a power car capable of handling a train with anywhere from 1 to 10 cars coupled, depending upon the nature of the service. It should be particularly emphasized that these cars were designed with sufficient strength, buffing and draft capacity to operate safely and satisfactorily with existing heavy equipment.

During this period, the primary incentive in substantially all cases was minimum overall cost to the railroads, so that in general these designs were made for minimum first cost consistent with the greatest possible reliability in service. . . . Most applications of cars of this type have been either on branch lines, secondary lines, or if operated on the main line, were in frequent stop, local service. They have, in practically all cases, been considered as a substitute, train for train, for the previous steam train equipment. . . .

It would now seem to be possible to define a new sort of application for motor-train equipment. Motor trains could be operated at advantage to the railway and to the traveling public if (1) the length of the journey is such that the operating of a personal car becomes somewhat of a task, (2) the speed and frequency of service are such that the time from point of origin to ultimate destination will be no more than with the private automobile, (3) the equipment is comfortable, safe and generally attractive, (4) the cost of such service is at least no greater, and (5) if there is a sufficient potential volume of traffic to justify such service.

Probably the above conditions can be met if (1) the distance is not too short (probably not less than 150 to 200 miles), (2) the equipment is operated primarily as a through service, avoiding the delays incident to secondary stops, thus permitting high schedule speed, (3) equipment designed for reasonable first cost and operating cost, and (4) if the traffic congestion on the railway does not too seriously interfere.

—Excerpts from a 1934 Semi-Annual Meeting paper on "Trends in the Design and Application of Motor Trains" by Charles O. Guernsey, The J. G. Brill Co.